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AND NAVAL ARCHITECT.

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JUNE, 1923.

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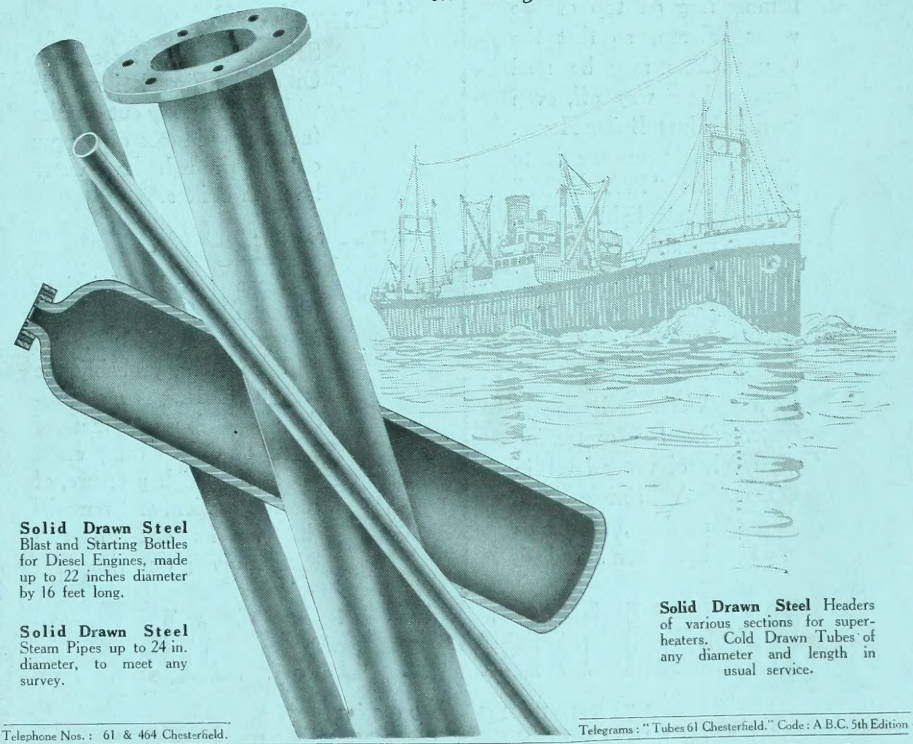

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The Editor will be glad to consider original articles and paragraphs on all subjects indicated by the title. Short practical articles based on the sender's own experience and illustrated by photographs or drawings are desired.

VOL. XLVI. No. 549. JUNE, 1923.

CONTENTS.

	PAGE.
PERSONAL PARAGRAPHS	197
EDITORIAL	198
Performance Standards for Merchant Ships—Marine Turbine Development—The Lubrication of Marine Engines—The "Titanic" Engineering Staff Memorial Fund.	
BOOKS FOR THE MARINE ENGINEER AND NAVAL ARCHITECT	200
PROPELLERS FOR COMMERCIAL VESSELS (Illus.)	201
COASTAL MOTOR TANK SHIP (Illus.)	203
MODERN MACHINERY SPACES—IV. (Illus.)	205
THE PROPELLING MACHINERY OF PLEASURE STEAMERS	216
TEXT-BOOK ERRORS IN MARINE BOILER WORK (Illus.)	217
A NEW SPANISH MOTOR SHIP (Illus.)	220
THERMAL AND MECHANICAL EFFICIENCIES IN STEAM AND MOTOR SHIPS—II (Illus.)	221
THE "BRITISH KROMHOUT" MARINE OIL ENGINE (Illus.)	225
REFRIGERATOR BARGES	227
THE NEW BRITISH INDIA MOTOR LINER "DALGOMA" (Illus.)	228
A NEW MOTOR LIFEBOAT	228
THE CLOSED FEED SYSTEM (Illus.)	231
MARINE DIESEL ENGINES—RUNNING CONDITIONS AND OPERATING SYMPTOMS—IV	233
NOTES FROM THE FOREIGN PRESS	235
RECENT PATENTS RELATING TO MARINE ENGINEERING	237
OFFICIAL NOTICES	238

Mr. G. L. Leeuwenburg, Inspector for Holland of the Bureau Veritas, died recently at Wassenaar at the age of 47 years.

Mr. Q. J. Terpstra, Secretary of the Dutch Association of Employers, The Hague, has been appointed Managing Director of Wilton's Dock and Shipyard Co., Rotterdam.

Mr. T. Macmillan has left the service of Barclay, Curle & Co. Ltd., on his appointment as foreman engineer on repair work with Cammell Laird & Co. Ltd., Birkenhead.

Col. John M. Denny, C.B., D.L., Chairman of Wm. Denny & Bros. Ltd., shipbuilders, Dumbarton, who died in December last in his sixty-fifth year, left estate valued at £95,811.

Herr Konteradmiral v. Gohren, formerly in charge of the Marineleitung, Hamburg, died in that city recently; his interests embraced the mercantile marine as well as the navy, and to both he rendered valuable services.

Commander Charles Worthington Craven, R.N., O.B.E., of Vickers Ltd., Barrow-in-Furness, has been appointed Deputy to the Managing Director, Sir James McKechnie, with full charge of the Barrow works during his absence.

The late Mr. Henry Robert Champness, M.V.O., Naval Architect, formerly Chief Constructor at Malta Dockyard, has left estate of the gross value of £7,935, the net personalty being £6,628.

At the annual meeting of the Liverpool Engineering Society held recently, Mr. John A. Brodie, M.Eng. (Liverpool), M.Inst.C.E., M.I.Mech.E., was elected to succeed Professor T. B. Abell as President for the ensuing year.

Mr. David Brownlie, of 46, Grange Road, Ealing, London, W.5, has, by amicable arrangement, severed his connection entirely with Brownlie & Green Ltd., of Manchester and London, and is now engaged in private practice as an advisory specialist in all matters relating to fuel economy.

At the general monthly meeting of the members of the Royal Institution, held on May 7, the President, the Duke of Northumberland, in the chair, Sir J. J. Thompson was elected Honorary Professor of Natural Philosophy, and Sir Ernest Rutherford, Professor of Natural Philosophy.

Mr. Stanley S. Cook, B.A., M.I.N.A., of the Parsons Marine Steam Turbine Company, delivered, on Monday, May 14, the last of the series of three Howard Lectures before the Royal Society of Arts, at John Street, Adelphi, W.C.2. The subject dealt with was "The Development of the Steam Turbine."

On Tuesday, May 15, the Duke of Northumberland, as President of the Institution of Naval Architects, and the Duchess of Northumberland, gave a luncheon at Syon House, Brentford, to members of the Institution. About 500 guests attended the "At home" which was held afterwards.

Mr. J. J. Domville Botterell, senior Partner and Founder of the firm of Botterell & Roche, Solicitors, died on May 12, at his office. Mr. Botterell is the third well-known City Solicitor with an extensive shipping practice to pass away within the last few months, the other two well-known names, which will be recalled, being Sir William Crump and Mr. Garrett.

The death has taken place at Southampton of Sir James Lemon, who was a Past-President of the Institution of Municipal and County Engineers and a Fellow of the Sanitary Institute. Sir James, who was in his ninety-first year, was twice Mayor of Southampton, and during the period of office, which was in 1891-2 and 1892-3, he saw the coming of the American Line to Southampton.

Mr. Charles Ernest Langtry, Assistant Repairs Manager with Cammell Laird & Co. at Birkenhead, has left the service of that firm after a connection extending over thirty-six years. Educated at Birkenhead School, Mr. Langtry served his time at Birkenhead Ironworks, commencing in 1887, and after seagoing experience in the American Line returned to Birkenhead to take up a position in the engineering department. His grandfather founded the concern known as Langtry's Line, in Belfast, in 1819, a concern taken over by the present Belfast Steamship Co. in 1859.

The Council of the Institution of Civil Engineers have made the following awards in respect of papers read and discussed at the ordinary meetings during the session 1922-1923:—Telford Medals to Mr. H. W. H. Richards (London) and Mr. E. O. Forster Brown (London); A George Stephenson Medal to Mr. Asa Binns, M.Inst.C.E. (London); a Watt Medal to Mr. A. B. Buckley, Jun., O.B.E., Assoc.M.Inst.C.E. (Winchester); Telford Premiums to Mr. W. A. Fraser, M.Inst.C.E. (Edinburgh), Mr. S. L. Rothery, Assoc.M.Inst.C.E. (Calexico, U.S.A.), Mr. Mark Randall (Johannesburg), and Mr. D. E. Lloyd-Davies, M.Inst.C.E. (Cape Town); an Indian Premium to Mr. D. H. Remfrey, Assoc.M.Inst.C.E. (Calcutta); a Manby Premium to Mr. F. M. G. Du-Plat-Taylor, M.Inst.C.E. (London); and a Crampton Prize to Mr. F. W. Jameson (Kimberley).

EDITORIAL.

PERFORMANCE STANDARDS FOR MERCHANT SHIPS.

SOME further interesting work, with the object of establishing standards of performance of the active vessels of the United States Shipping Board Fleet, has been carried out by the Fuel Conservation Committee appointed by the Vice-President of the Board. The latest results are worth noting by those interested in the economic running of steam propelling machinery on cargo vessels, especially since the vessels concerned are owned and operated by the Shipping Board. The first set of standards established was for the 502-ft. ships, and the principal results were made public towards the end of last year. The work has now been extended to include the 535-ft. passenger-cargo vessels, and the pure cargo vessels known generally as the Hog Island ships. The 535-ft. passenger and cargo vessels have been placed in three groups, and standard curves have been prepared for each group for varying draughts showing on a base of speed in knots the theoretical number of nautical miles covered per ton of fuel oil consumed. Then by comparing actual results with them a general average performance for the class has been tabulated as shown below :—

Draught in ft.	R.P.M.	Observed speed kts.	Fuel consumption.		Nautical miles per ton.		
			Tons per 24 hrs.	Pounds per mile.	A.—Actual.	B.—Theoretical.	Percentage efficiency.
24′ 5 11/16″	113·9	16·9	133	725	3·08	3·54 From Curves.	87

The percentage efficiency, it will be noted, is determined by a comparison of the actual nautical miles covered per ton of fuel with the theoretical figure as determined from the curves. For the class the performance efficiency varies considerably with individual ships, being as low as 62·7 per cent. in one vessel and as high as over 90 per cent. in eight of the sixteen. The results for the Hog Island cargo vessels, however, show a much wider range of efficiency—a maximum of 107·8 per cent. in one case and a minimum in another of 58·7 per cent., the average for the entire class being 83·67 per cent., with details as shown below :—

Draught in ft.	R.P.M.	Observed speed kts.	Fuel consumption.		Nautical miles per ton.		
			Tons per 24 hrs.	Pounds per mile.	A.—Actual.	B.—Theoretical.	Percentage.
21′ 8 7/16″	80·6	10·33	29·14	26·33	8·51	10·17	83·67

It is interesting to note from this that the average actual consumption is considerably less than the theoretical consumption, but at the same time it seems strange that there should be such a wide range of efficiency in vessels of the same type. The vacuum system was tight in one case apparently, and the vessel developed full power on a low fuel consumption ; with a leaky system in another case the consumption showed a corresponding increase. The curves themselves were prepared partly from theoretical considerations, but they do not represent impracticable values. In fact, they can be considered rather as values obtained by several vessels of the class under consideration with both machinery and engine-room staff working efficiently. The results are both valuable and interesting ; they give to the marine engineer some data as to the actual performance of oil-fired ships, and they form an interesting comparison of the difference in working between similar ships—sister ships—belonging to the same fleet. The whole scheme represents, in fact, the virtual establishment of an ideal for a class, and then a determination of how nearly the ideal is approached by individual vessels of the class. The results are instructive also, in that they establish some definite performances of oil fuel consumption when burnt in ships' boilers and they are useful to form a comparison with similar figures for coal-fired boilers and also for internal-combustion engines. Such figures are useful possessions for those concerned with the operation of a fleet of similar vessels, and serve as a possible indication of the weaknesses in the working of the propelling plant on individual ships.

MARINE TURBINE DEVELOPMENT.

THE marine steam turbine has been discussed widely in technical circles of recent years, and its value debated and compared with other types of available prime movers for propulsive purposes. The Howard Lectures for the current session of the Royal Society of Arts delivered by Mr. Stanley S. Cook, B.A., of the Parsons Marine Steam Turbine Company, dealt with the development of the steam turbine and were remarkably valuable in that they showed the immense strides which have been made not only in marine work but also in land installations. The whole subject was reviewed in three lectures, the first two of which dealt more especially with marine work and the last of which concentrated principally on land work. The trouble experienced in the adoption of the direct-coupled turbine for low speed vessels was touched upon, and the impossibility of combining best speed of turbine with most efficient propeller speed. The author also mentioned the well-known arrangement consisting of a combination of reciprocating engines and turbine, the turbine driving the centre shaft and the reciprocating engines the wing shafts; the latest exponent of this arrangement is, of course, the Red Star liner *Belgenland*, whose machinery we describe in detail elsewhere in these pages. The need for mechanical gearing and its introduction was also discussed, and mention was made of the now almost classical experiments which were carried out in the old *Vespasian*, whose original geared turbines, it is interesting to note, are still running in the steamship *Lord Byron*. Types of gearing were discussed, and special emphasis was laid on the advantage of the hunting tooth. The Michell thrust was considered in detail and its advantages over the ordinary multi-collar type of thrust block, and the author also referred to some interesting pressure experiments which were carried out in connection with the Michell thrust. Methods of blade attachment were dealt with very fully, and a comparison was drawn between the efficiencies of installations now attainable and those attainable even 10 years ago. Double-reduction gears were discussed, and we called to mind the author's defence of reduction gearing from a metallurgical point of view, written in collaboration with Sir Charles Parsons, and read before the Spring Meetings of the Institution of Naval Architects in March last. Two methods of wheel and pinion arrangement were mentioned, and the three-stage turbine arrangement adopted in the *Somerset* was alluded to ; as showing a further stage of development, the author mentioned also the advantages derived from the underslung condenser, the arrangement adopted wherever possible in present-day practice. The three lectures, while of course containing very little of a controversial nature, were extremely valuable as an indication of the progress that has been made in turbine design during recent years, and they left one with a strong impression that the *dernier mot* of the turbine has by no means been pronounced as yet.

THE LUBRICATION OF MARINE ENGINES.

ONE of the most important details in the design and construction of a marine engine is the lubrication system, and the changes which have taken place in the types of engine utilised for the propulsion of ships have tended to give an added importance to this detail; to-day, in fact, it is more than ever necessary that the greatest care and attention should be given to the means whereby the lubricant is supplied to the various parts of the machinery, and also in the choice of a proper lubricant. The tendency in the design of marine engines has been in the direction of using steam of higher pressures and therefore of higher temperatures, while as well as having a high temperature the steam is frequently superheated. Again, the introduction of the geared turbine has led to the employment of higher speeds, which in its turn demands more efficient methods of lubrication of the bearings, while if we consider the case when internal-combustion engines are employed, not only are we faced with machinery in which very high temperatures are employed, but the speed is very often high and the maximum pressure on the piston and therefore on the bearings is higher than is the case of the reciprocating steam engine. Of particular importance is internal lubrication in modern engines, *i.e.*, lubrication of pistons. When wet steam is employed—and the steam may always be considered wet in the absence of superheating, even if it is dry saturated on leaving the boiler—the presence of moisture acts as an efficient lubricant, but when the steam is superheated it becomes necessary to employ some form of internal lubrication. For this purpose only the best mineral oil should be employed, and this may be carried into the engine with the steam, the oil being continuously fed into the system, say, by means of a sight-feed lubricator. In addition to the liquid oils, various lubricating greases are used, often in combination with a certain proportion of graphite. Graphite alone may be used, and its lubricating qualities are of the highest order, since it possesses the property, especially with cast iron, of filling the pores of the iron, thus forming a kind of graphite metal skin on the surface which possesses a very low coefficient of friction. The finely divided graphite is forced into the steam chest as a spray by a small jet of steam, the apparatus resembling an injector in principle. With internal-combustion engines, internal lubrication is generally provided directly through a pipe which is led direct to the cylinder walls at about mid-stroke.

Of course, continuous lubrication is really the ideal system, and this can be adequately provided for when forced lubrication is employed. The idea is not a new one, it having been successfully used for many years in dynamo and other high-speed engines, and whilst it is not employed to any large extent on reciprocating marine engines it is the general practice with turbines and internal-combustion engines. Superior efficiency and economy of oil are the principal claims which are urged for the forced system of lubrication over other oiling arrangements, and both claims have been established beyond question by installations now in operation. With forced lubrication the oil is used over and over again and not wasted in the bilge after performing its function but once, and the oil consumption, when forced lubrication is employed, may be taken as from five to eight pounds per 2,000 i.h.p. per hour, thus showing a saving of lubricating oil of at least 100 per cent. The superior efficiency is, however, of greater importance than superior economy, and the former is due to the fact that by employing oil under pressure we can be assured that there always exists between the shaft or pin and its bearing a thin film of oil, so that the metal surfaces are never actually in contact. The moving part is thus "oil borne," and with a good lubricant this means a very low coefficient of friction with a resulting increase in the mechanical efficiency of the engine.

THE "TITANIC" ENGINEERING STAFF MEMORIAL FUND.

WE learn that the above Fund, administered by the Institute of Marine Engineers, needs more money. The war having intervened so soon after its inception, it may be that the circumstances in which the Memorial Fund was founded have become somewhat dim in the memories of some of us. It will be recalled that, following the disaster to the *Titanic*, a world-wide expression of appreciation of the heroic behaviour of the engineering staff of the ship, the members of which were all lost, resulted in a fund being subscribed towards the erection of a permanent memorial. The total amounted to £2,800, of which £300 was devoted to the erection of a memorial at Southampton, £50 to a memorial tablet in the premises of the Institute of Marine Engineers, while the balance was invested to provide a benevolent fund in perpetuity to assist the dependants of engineers lost at sea.

It will be regrettable if the good work so well begun should be seriously limited by want of funds. The more recent losses of the *Egypt* and others remind us that many an engineer meets his death at sea in similar circumstances, and it is in such cases that the *Titanic* Memorial Fund affords timely help. We recommend it to the sympathy of those who can assist. Donations should be sent to the Hon. Secretary, Institute of Marine Engineers, 85, Minories, London.

From the "Marine Engineer," June, 1879.

THE DILLON SYSTEM OF RAISING SUNKEN VESSELS.—On Thursday morning, the 22nd May, His Royal Highness the Duke of Edinburgh honoured Mr. Thomas A. Dillon with a visit at his rooms, at the Westminster Palace Hotel, to inspect the latest development in his system of raising sunken vessels by the aid of compressed air. . . . The invention consists in a method of causing sunken vessels to rise and float without the aid of divers, and without in any way touching the ship. This is effected by certain pneumatic and hydrostatic laws, which have hitherto been either unobserved or neglected, but which Mr. Dillon has laid under contribution in the attainment of his desired end.

DEATH OF MR. WILLIAM FROUDE.—We regret to announce the death of Mr. William Froude, F.R.S. The deceased gentleman, who had long been in a weak state of health, left England in November last year on board Her Majesty's ship *Boadicea* for a holiday cruise to the Cape, and he died of dysentery on the 4th inst., at Simon's Town. Mr. Froude, who was one of the greatest masters of applied mathematics in modern times, served on Lord Dufferin's Committee on the designs upon which ships of war have been constructed, and also upon the committee which was appointed to examine and report upon the stability of the *Inflexible* turret-ship.

STEEL IN SHIPBUILDING.—The past month, like that which preceded it, has been marked by the assembling of one of those important institutions which combine science with industrial pursuits. Now that iron has largely supplanted wood in shipbuilding, to be supplanted, to all appearance, in its turn by steel, what passes in the meetings of the Iron and Steel Institute must be of the utmost interest to the marine engineers. Two papers directly bearing upon shipbuilding material were read at the late meeting of that society. That of Mr. Adamson, the well-known pioneer of the use of steel in the construction of steam boilers, had its value, although in the combined discussion on these two essays, with that of Mr. H. R. Maynard on the use of steel in bridges, Mr. Barnaby's on the use of the same material in naval architecture almost wholly engrossed the attention of the meeting. The strength and endurance of steel as compared with even superior iron must eventually bring it into very extensive if not into general use.

BOOKS FOR THE MARINE ENGINEER AND NAVAL ARCHITECT.

Brief Reviews of Recent Publications.

Entwurf und Einrichtung von Handelsschiffen. By Dipl.-Ing. Prof. Heinrich Herner. Leipzig: Dr. Max Jänecke, Hospitalstrasse 10, Third enlarged edition; 5½ in. × 8¼ in. × ⅜ in.; Diagrams, sketches and folding plates. Price 12s. net.

The design and equipment of mercantile vessels is a vast subject to compress into so small a space, but the author of this book has contrived to deal with it very thoroughly. He considers ships as a whole, briefly, fundamentally, historically, and then goes on to deal with the more detailed portions of ships' structures. A complete section is devoted to sailing ships and their fittings—the Germans seem ever to have a "soft spot" for this type of vessel—and the full-page drawings containing little sketch elevations of typical sailing craft are instructive and interesting. The author also deals in another section with the important question of tonnage and tonnage measurement. We do not pretend to admire the arrangement of the folding plates at the end of the book, however. They are arranged to slip into a thin paper band, which, under stress of even occasional wear, cannot possibly have a very long life.

Electric Cranes and Hauling Machines. By F. E. Chilton, A.M.I.E.E. Pitman's Technical Primers. London: Sir Isaac Pitman & Sons Limited, Parker Street, Kingsway, W.C.2. 4¼ in. × 6½ in. × ⅜ in. Price 2s. 6d. net.

This is one of the recent productions of the Pitman Technical Primer series, and it deals with the construction and operation of modern cranes, transporters, capstans, winches, haulages, winding gear, &c., and their principal accessories. This is a large field to be covered by so small a book, but, as the author himself specifically states in the preface, "The subject is treated in a simple and descriptive manner on broad and general lines. For the reader who is desirous of further information there are several large works which deal very thoroughly with the many ramifications of the subject." Prospective readers in shipping circles will find Chapter VII, which deals with electric capstans and electric ship winches, quite interesting, and engineers generally will find the book worth careful perusal.

The Motor Ship Engineer's Log. (Pattern No. 115.) Published by James Brown & Son, 52 to 58, Darnley Street, Glasgow. 12¼ in. × 9¾ in. × ⅛ in.

This little book is issued for monthly or three-monthly use, and it is intended for watchkeeping engineers in motor ships. Each sheet of the book contains columns subdivided for the complete 24 hours of watches, revolutions of engines, &c. There are also columns for recording particulars of the various oils used, the weather conditions and the number of hours the machinery is in use.

Pattern Making. By Edward M. McCracken and Charles H. Sampson, B.S. London: Scott, Greenwood & Son, 8, Broadway, Ludgate Hill, E.C.4. 11½ in. × 8¼ in. × ½ in. 120 pp. and 30 plates. Price 10s. 6d. net.

The authors of this book are heads respectively of the Woodworking Department, Johnson School, Scranton, Pa., and of the Technical Department, Huntingdon School, Boston, Mass. The object of their book is "to fill the need for a text, the use of which will give to students such a knowledge of the principles of pattern making as to enable them to make any pattern of the usual type," and in the space the authors have at their command the ground is quite well covered. The first section deals with general methods of pattern making, and this is followed by one on pattern details. The section dealing with hand tools is well illustrated, as is also that dealing with pattern makers' machines. The book is illustrated both with plates and with line drawings, and should make quite useful reading to the student of engineering.

The Internal-Combustion Engine. Volume II. By Harry R. Ricardo, B.A., A.M.I.C.E., M.I.A.E. London: Blackie & Son Ltd., 53, Old Bailey. 7 in. × 10½ in. × 1⅜ in. 373 pp. Price 30s. net.

This is the concluding volume of Mr. Ricardo's work on internal-combustion engineering, and it deals mainly with high-speed engines. In an excellent introduction the author points out the present position of the light internal-combustion and its limitations; he also touches on the somewhat disastrous effects of a world shortage of fuel for such engines. The book contains a mass of useful information; road engines, aero engines and tank engines are dealt with, and the diagrams and plates are numerous and well produced.

The Directory of Shipowners, Shipbuilders and Marine Engineers, 1923. Compiled under the direction of the Editor of *Shipbuilding and Shipping Record*. London: The Directory Publishing Co. Ltd., 33, Tothill Street. 8½ in. × 5¼ in. × 1¼ in. 728 pp.

The 21st edition, which has just been published, has a great many more entries than its predecessors, but, owing to some slight modifications in the arrangement, the number of pages has been reduced. So many people use this directory when travelling that the reduced bulk will be welcome. It now measures only 8½ in. × 5¼ in. × 1¼ in., and in some 730 pages contrives to give something like 25,000 entries. The indexes have been compiled with even greater care than usual, with extensive cross references. One of the most valuable features of the book is what one may term the official side—that is, the names of the directors, secretaries, marine superintendents and other officials of shipping companies, and in the shipbuilding section the managers, chief draughtsmen, and others. The fleets of the shipping companies are given, together with such information as gross, net and deadweight tonnage, passenger accommodation, speed, whether oil-fuel burning, and so on. In the shipbuilding section the capacities of the shipyards and engine works are stated, and the dimensions of dry docks where ship-repairing is carried on. Other information includes a list of societies, institutions and trade and labour federations, and a directory of consulting naval architects and marine engineers. A most useful feature of the work is the large amount of information concerning foreign companies.

TRADE PUBLICATIONS.

Adam Hilger Limited, 75A, Camden Road, N.W.1, have issued a small book descriptive of Professor Coker's apparatus for determining the distribution of stress in structural and machine members. The information contained in the book is very useful.

Ronald Trist & Co. Ltd., 12, Clipstone Street, Great Portland Street, W.1, have just issued a little folder descriptive of their patent "Raybeam" electric gauge glass indicator. This is a remarkably ingenious little device, and by means of a reflected light indicates the exact position of the top of the water column in the gauge glass.

PROPELLERS FOR COMMERCIAL VESSELS.

Graphic Chart for Design—Analysis—Wake Values—Slips—Thrusts. (From Froude's Curves.)

The chart reproduced on the next page, giving Froude's values as deduced from his model experiments, has been prepared to combine in a single diagram a means of working out any of the usual propeller problems met with in the ordinary course of design, analysis of trial trip data, &c., in a quick, simple and direct manner, and in what follows a knowledge of Barnaby's, Froude's, Taylor's and Baker's works is assumed.

The diagram can be used with Froude's characteristic coefficients, C_A and C_o, which were devised by him for design.

$$\left(C_A = \frac{R^2H}{BV_1^5}\right) \text{ and } \left(C_o = \frac{H}{BD^2\,V_1^3}\right)$$

but it is found simpler to use the coefficients

$$C_A \text{ and } \sigma \left(= \frac{V_1}{DR}\right) \text{ for design.}$$

The following symbols will be used :—

H = thrust horse-power.
S.H.P. = shaft horse-power.
D = diameter of propeller in feet.
V_1 = speed of propeller in knots per hour through wake water.
V = speed of ship.
B = Froude's blade factor.
R = revolutions per minute in hundreds.
T = thrust in pounds.
p = analysis pitch ratio.
s = slip ratio.

For analysis of trial trip data it is essential first to find V_1. This can be found conveniently from the characteristic coefficients—

$$r = \frac{H}{BD^2R^3}, \text{ and } \sigma,$$

in the following manner.

The diagram is logarithmic, and, to use it, draw a line from the value of the coefficient on the appropriate scale at right angles to it. Any spot on this line will satisfy the value of the coefficient. Consequently, the point of intersection of two such lines will satisfy two of the coefficients.

Notice $r = \frac{H}{BD^2R^3}$ does not contain V_1.

As we do not yet know H, we must adopt an artifice, and, instead of r, use $r^1 = \frac{S.H.P.}{BD^2R^3}$.

For reciprocating engines the S.H.P. may be calculated from the I.H.P., and B, D and R are known.

Draw a normal to the r scale through r^1.

Draw the constant p (analysis pitch ratio) line appropriate to the propeller in question.

This line will represent the $\frac{H}{BD^2V_1^3}$ on $\frac{V_1}{DR}$.

From this line deduce another line to represent the $\frac{S.H.P.}{BD^2V_1^3}$ on $\frac{V_1}{DR}$.

This is done by taking several convenient points on the p line, and as

$$\frac{S.H.P.}{BD^2V_1^3} = \left(\frac{H}{eBD^2V_1^3} = C^1, \text{ say}\right)$$

where e = the propeller efficiency.

Reading the values of C_o at these points, we divide it by the values of e, which are found from the constant e curves, and obtain values of C^1, which are plotted on the various values of σ, and a curve drawn through them.

The normal through r^1 cuts this "C^1 on σ curve" in a point which solves the problem.

Referred to the σ scale, σ is obtained, and $V_1 = \sigma$ D R.

The normal through σ cuts the p curve in a spot from which follow values of r, C_o, C_A and efficiency, from any of which H can be calculated and slip is got from σ as

$$s = \left\{1 - \frac{1{\cdot}0133\,\sigma}{p}\right\},$$

or it can be got from the diagram by stepping back from σ an amount equal to the logarithm of the pitch ratio, and taking a point opposite this on the slip scale (as indicated by the example in the diagram).

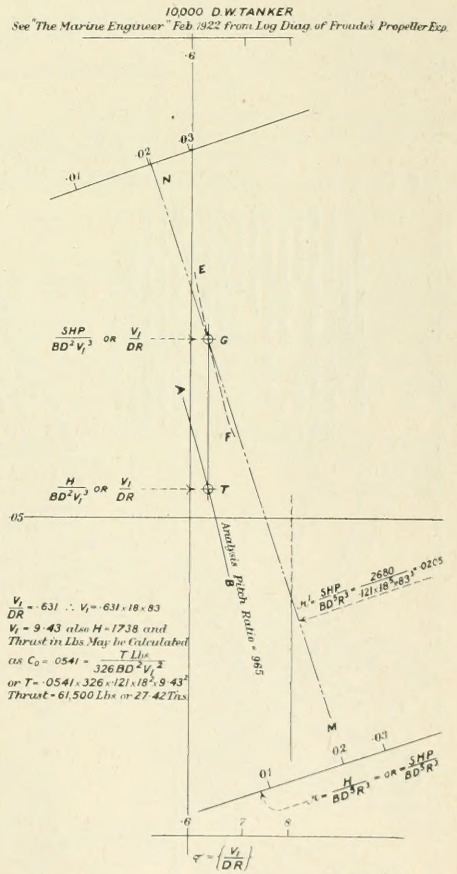


Diagram illustrating Procedure Adopted in a Specific Case.

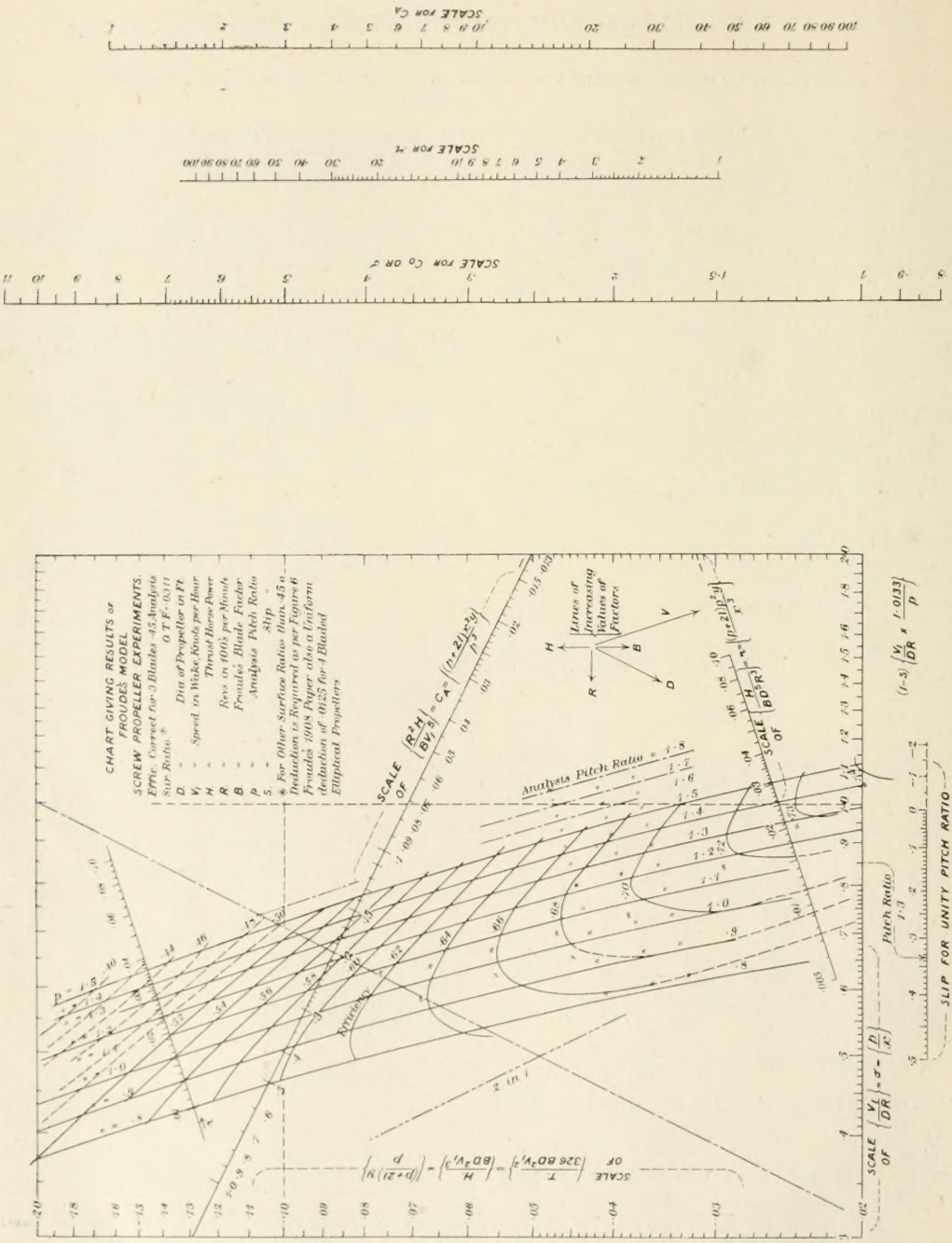


CHART GIVING RESULTS OF FROUDES MODEL SCREW PROPELLER EXPERIMENTS.
Effic. Correct for 3 Blades 45 Analysis
Sur Ratio.*
D. = Dia of Propeller in Ft.
V1 = Speed in Wake Knots per Hour
H = Thrust Horse Power
R = Revs in 000s per Minute
B = Froudes Blade Factor
P = Analysis Pitch Ratio
S = Slip
* For Other Surface Ratios than 45 a Deduction is Required as per Figures 6 Froudes 1908 Paper also a Uniform deduction of 0025 for 4 Bladed Elliptical Propellers
SCALE OF $\left(\frac{R^2H}{BV_1^5}\right) = C_A = \frac{(n+2)(P+2)^{n-2}g}{P^{n-2}g}$
Lines of Increasing Values of Factors
H
V
R
B
D
Analysis Pitch Ratio = 1·8
1·7
1·6
1·5
1·4
1·3
1·2
1·1
1·0
·9
·8
SCALE OF $\left(\frac{H}{BD^2R^3}\right) = \frac{(n+2)(P+2)^{n-2}g}{P^{n-2}g}$
SCALE OF $\left(\frac{V_1}{DR}\right) = \sigma = \left(\frac{P}{\pi}\right)$
Pitch Ratio
SCALE OF $\left(\frac{T}{326\,BD^2V_1^2}\right) = \left(\frac{H}{BD^2V_1^3}\right) = \left(\frac{(P+2)}{P}\right)^{n-2}$
$(1-S)\left(\frac{V_1}{DR} \times \frac{1·013}{P}\right)$
SLIP FOR UNITY PITCH RATIO
SCALE FOR C_A
SCALE FOR π
SCALE FOR C_0 OR σ

Logarithmic Propeller Chart prepared for working out Propeller Problems.

Taylor's wake fraction $= \left(\frac{V - V_1}{V}\right)$.

Note, any spot on the diagram gives instantaneous values of—

Analysis pitch ratio, efficiency, slip, $\left\{\frac{V_1}{DR}\right\}$, $\left\{\frac{H}{BD^2V_1^3}\right\}$, $\left\{\frac{R^2H}{BV_1^5}\right\}$ and $\left\{\frac{H}{BD^5R^3}\right\}$, if referred to the p line, but if referred to the "C^1 on σ curve," as developed above, it gives values of—

$$\left\{\frac{S.H.P.}{BD^2V_1^3}\right\}, \left\{\frac{R^2S.H.P.}{BV_1^5}\right\} \text{ and } \left\{\frac{S.H.P.}{BD^5R^3}\right\}.$$

Only one such curve is required for each propeller.

To find V_1 if actual thrust of propeller has been measured, proceed as follows :—

Use the $\left(C_o = \frac{T}{326\ BD^2V_1^2}\right)$, and σ scales.

First get a provisional spot on chart representing $C^1\left(=\frac{T}{326\ BD^2V^2}\right)$ and $\sigma^1\left(=\frac{V}{RD}\right)$, where V, the speed of ship, is used instead of V_1.

Through this spot draw a line at an angle of + 2 in —1 (or whose tangent is —2), cutting the p line in a point which will give values of σ and C_o, from which $V_1 = \sigma DR$.

Efficiency and slip are also read off.

The constant pitch ratio curves represent thrust on slip curves, for constant, V_1.

The constant p lines are spaced approximately logarithmetically, which is useful for purposes of interpolation.

Scales to be used with the chart are appended.

Example.—To find the speed of propeller through the wake water (V_1) from trial trip data.

Taking the published results of the 10,000 d.w. tanker (see the *Marine Engineer*, February, 1922).

I.H.P.	S.H.P.	Speed in knots.	Revs. per min.	R.
3,090	(say) 2,680	12	83	0·83

Hence $r^1 = \frac{S.H.P.}{BD^5R^3} = \frac{2,680}{0{\cdot}121 \cdot 18^5 \cdot 0{\cdot}83^3} = 0{\cdot}0205.$

(See diagram on page 201.)

Through r (in diagram) = 0·0205 draw MN at right angles to r scale. Taking analysis pitch ratio as ·965, or 1·12 times the face pitch ratio (see Abel's diagram), draw by interpolation the 0·965 line AB, and also draw the curve EF vertically above it an amount equal to the logarithm of the efficiency (an efficiency deduction of 0·0125 to be made in this case for four blades). A curve through 4 or 5 spots will be sufficient.

G, the point of intersection of MN and EF, gives the value of $\sigma = \frac{V_1}{DR} = {\cdot}631$.

Knowing D and R, $V_1 = 0{\cdot}631 \times 18 \times 0{\cdot}83 = 9{\cdot}43$ knots through wake.

Taylor's wake fraction $= \left\{\frac{V - V_1}{V}\right\} = \frac{12 - 9{\cdot}43}{12} = 0{\cdot}215.$

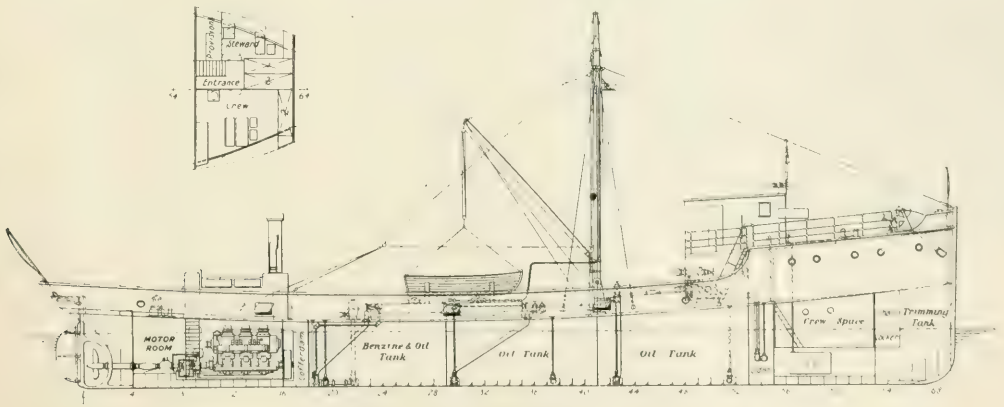
Thus V_1 is found by two calculations and a short graphic construction. This should be compared with the lengthy trial-and-error methods used ordinarily.

H = S.H.P. × propeller efficiency.

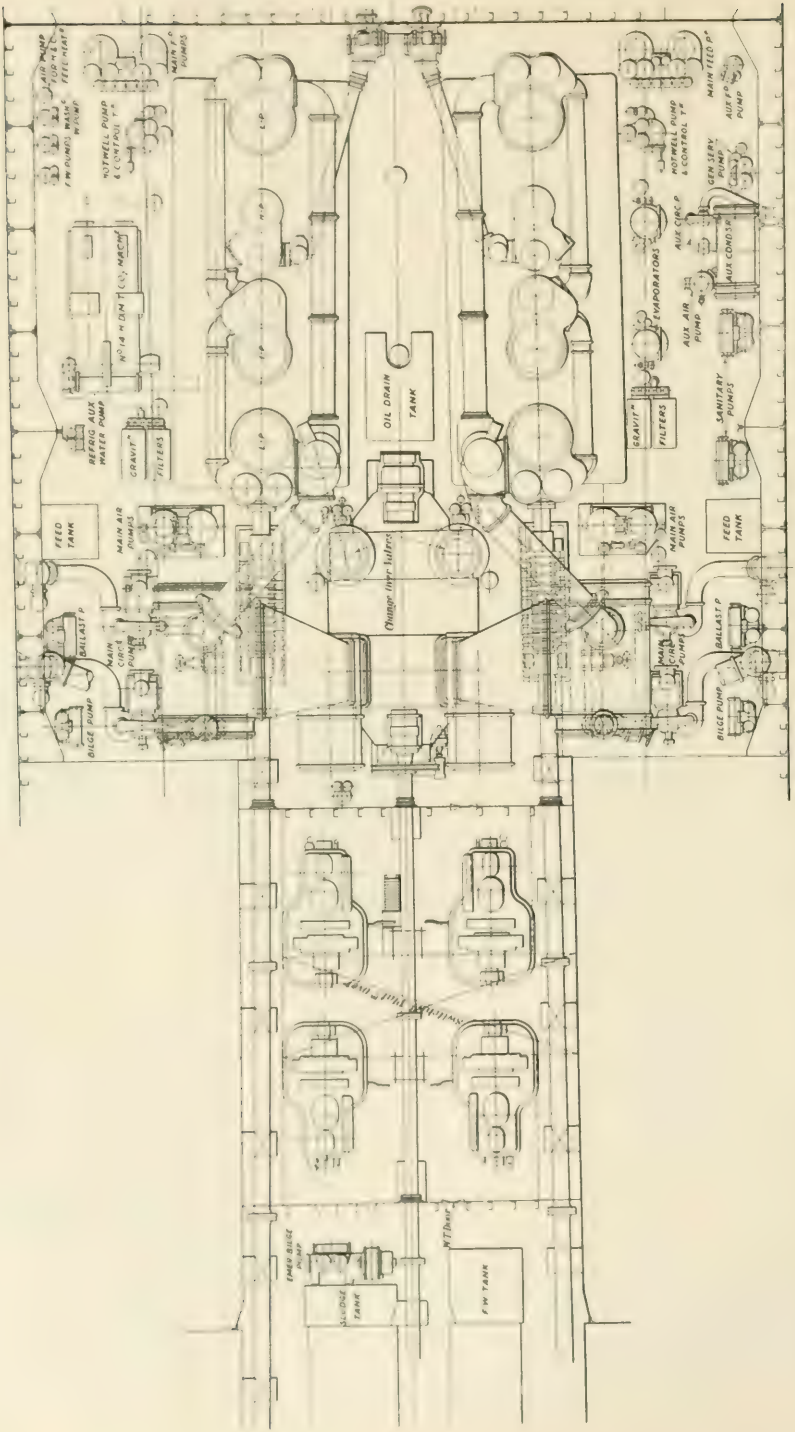
Coastal Motor Tank Ship.

Owing to the extended use of small motor-driven vessels, the demand for oil at small seaports has very largely increased during recent years. This state of affairs is particularly noticeable in the Baltic Sea where, at many of the small coast towns, a considerable demand for oil exists. Thus, in the case of a small fishing port with a fleet of 20 motor fishing vessels and about 10 other motor craft, aggregating perhaps 600 b.h.p., the monthly requirements will, at the lowest estimate, be from 36 to 40 tons of oil. Hitherto these supplies have had to be brought overland by means of tank wagons, and it was felt that it would lead to a considerable reduction in the price of oil if such supplies could be obtained from small motor-driven tank ships. The United Oil Company (Forenede Oljekompagnie), of Copenhagen, therefore approached the Danziger Werft, of Danzig, for the designs of a motor tank ship of 250 tons carrying capacity, to be driven by a crude oil engine of about 150 h.p. specially adapted to the work of supplying the small harbours of the Baltic with petroleum and other kinds of oil. As a result the *Bodil* has been put into service between Copenhagen and Fredericia. The principal dimensions of the *Bodil* are as follows :—

Length, overall	36·55 metres	(120 ft.)
Length, between perpendiculars	34·50 ,,	(113·2 ft.)
Breadth, moulded	7·20 ,,	(23·6 ft.)
Depth, moulded	2·60 ,,	(8·5 ft.)
Gross tonnage	203	
Net tonnage ..	95	



Profile of Motor Tank Ship "Bodil."

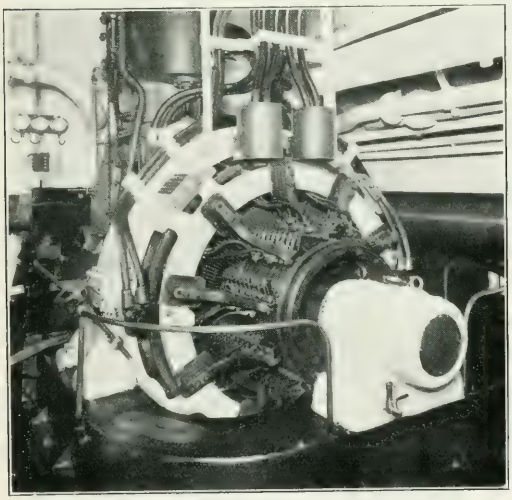


The Red Star Liner "Belgenland." Diagrammatic Plan of Machinery Space.

MODERN MACHINERY SPACES.—IV.

The Red Star Passenger and Cargo Liner "Belgenland," built and engined by Harland & Wolff Ltd., Belfast.

We describe below some features of the main and auxiliary machinery of the Red Star liner *Belgenland*. This vessel has had a somewhat eventful career: she was designed before the outbreak of hostilities in 1914, completed during the war period sufficiently to enable her to carry cargo, she was also fitted up for the carriage of troops, and now after refitting and a thorough overhaul, she has been placed on service between Antwerp and New York in her proper rôle as a passenger liner. The vessel was built by Harland & Wolff Limited at their Belfast yard, and the hull builders also supplied the propelling machinery and much of the auxiliary machinery. She left Belfast on March 17 bound for Antwerp,



One of the Main 300-kw. Generators.

and completed her maiden voyage towards the end of April. The following are the leading particulars of the vessel:—

Length, b.p.	670·4 ft.
Breadth moulded	78·4 ft.
Depth moulded	44·7 ft.
Draught moulded	36 ft. 3 in.
Deadweight	22,025 tons.
Gross registered tonnage	24,547
Net	15,440

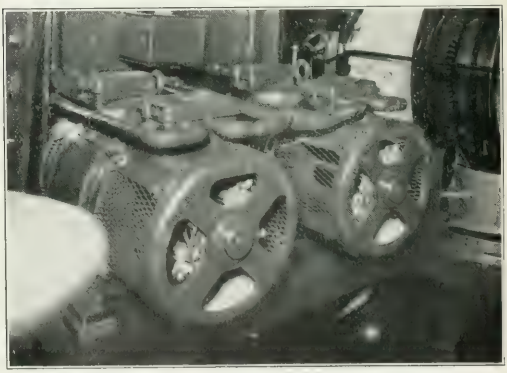
The *Belgenland* is one of the largest vessels trading between Antwerp and New York, and she has accommodation for approximately 500 first-class, 1,000 second-class, and over 1,000 third-class passengers. Eleven watertight bulkheads divide the vessel into twelve watertight compartments. The double bottom, which extends fore and aft, and the fore and aft peaks, are arranged for the carriage of water ballast. Electrically-operated watertight doors are arranged on the bulkheads, where access from one compartment to the next is absolutely necessary, and these are of the hull builder's own type. We illustrate on page 209 an example of these doors, the one shown being actually that leading from the starboard shaft tunnel into the tunnel flat aft. It is controlled by a Laurence Scott motor, which will be seen at the top of the door, giving 100 volts at 14 amps. and 530 r.p.m.

Main Propelling Machinery.

The arrangement of propelling machinery is one of which a very early exponent was the well-known Denny-built *Otaki* of

The Chief Engineer's Cabin.

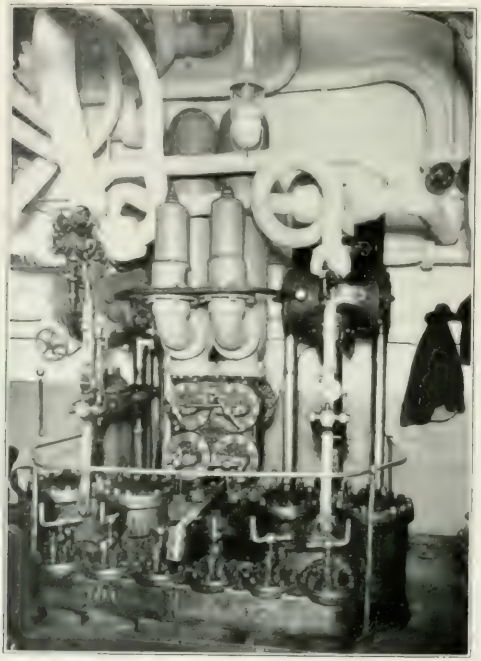
1908, namely, a three-shaft arrangement in which the two outer shafts are propelled by four-cylinder triple-expansion reciprocating engines, and the centre shaft by a direct-coupled low-pressure Parsons' turbine. Steam, after passing through its various stages of expansion in the cylinders of the two



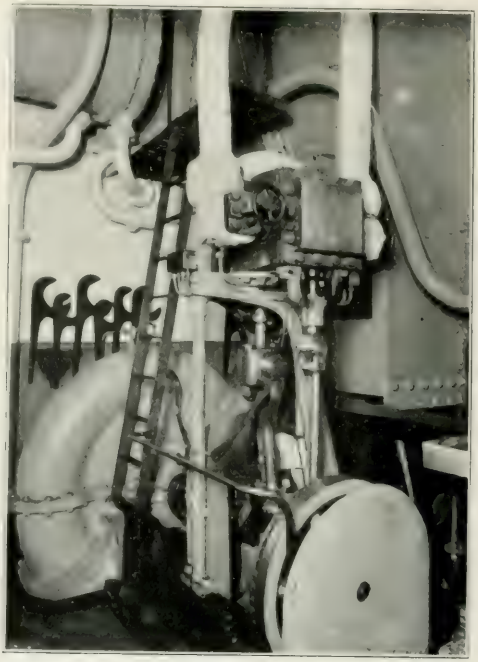
The High-Tension to Low-Tension Rotary Transformers for the Cumberland Electrolytic System.

reciprocating engines, is allowed, before passing into the condenser, to give up its remaining energy in the turbine. Alternatively, of course, when manœuvring for example, by means of the change-over valves, the port one of which we illustrate on page 210, the turbine may be cut out and the steam from

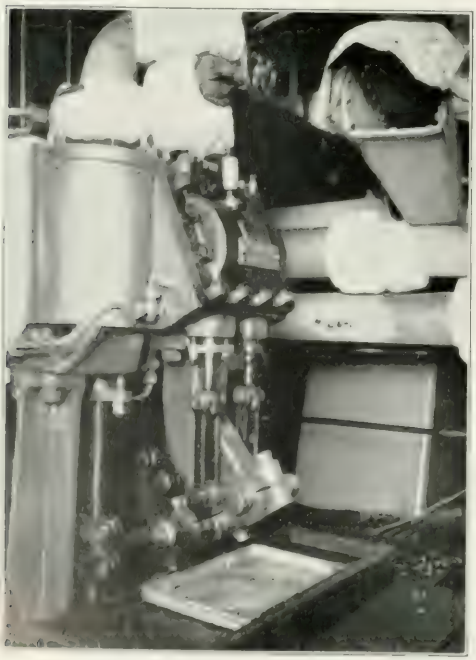
THE MACHINERY SPACE OF THE



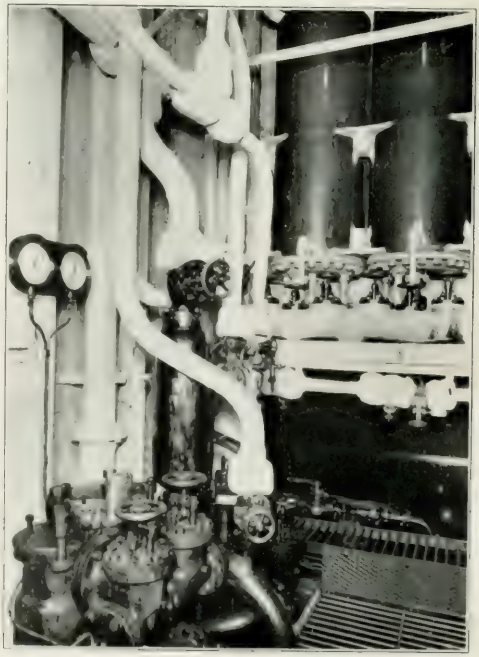
The Starboard Pair of Main Feed Pumps.



Main Circulating Pump, Starboard Side.



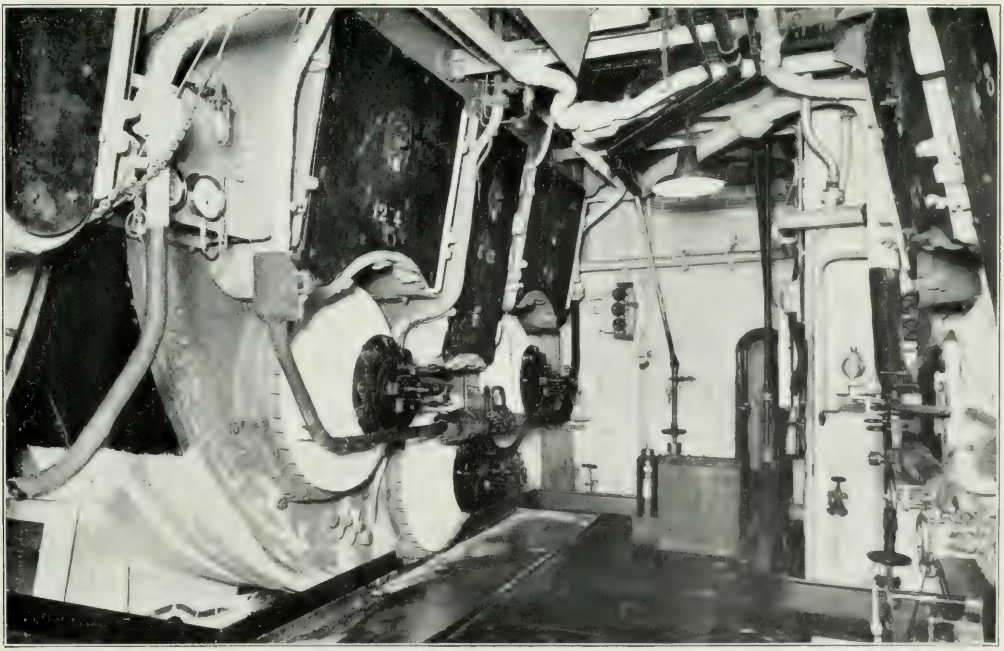
One of the Three Brine Circulating Pumps.



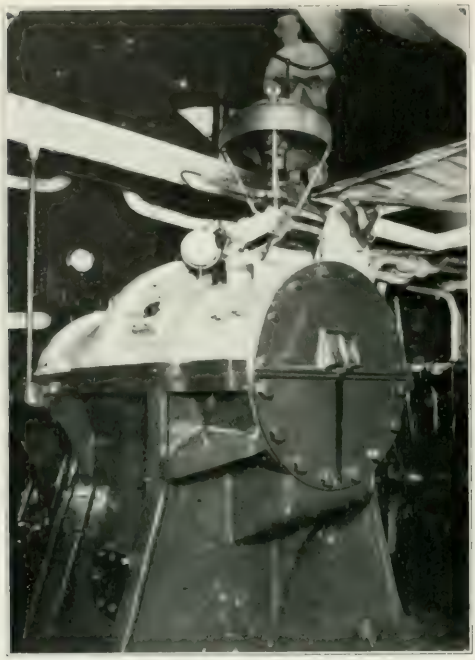
Part of the Starboard Forward Oil Fuel Unit.

RED STAR LINER "BELGENLAND."

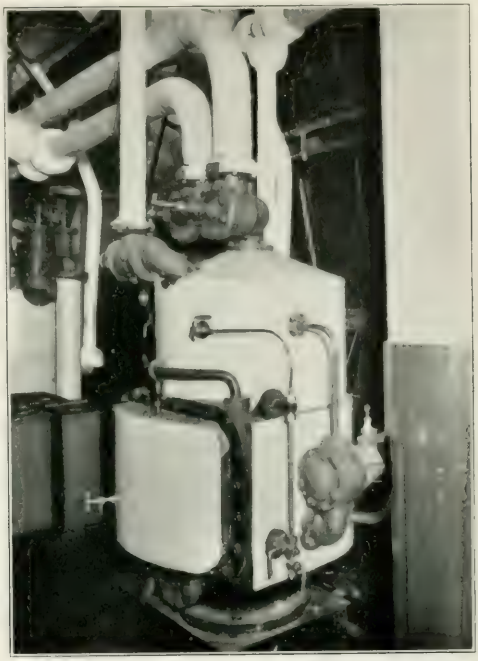
View showing Tube Cleaning on No. 8 Boiler. Removing the Retarders.



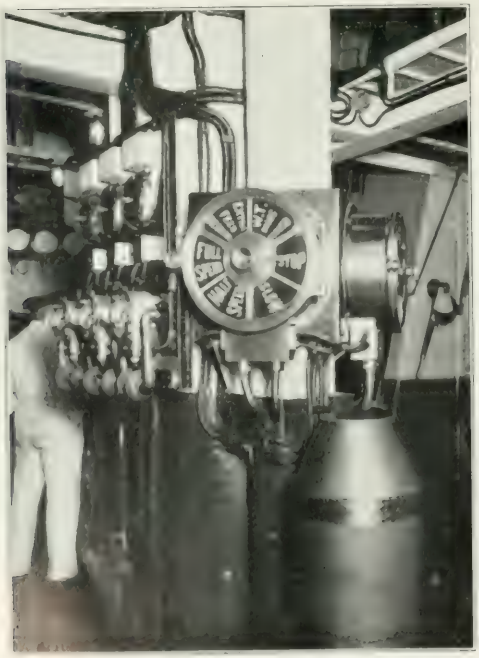
View of No. 6 Boiler of No. 5 Stokehold, showing the White Oil Fuel Installation.



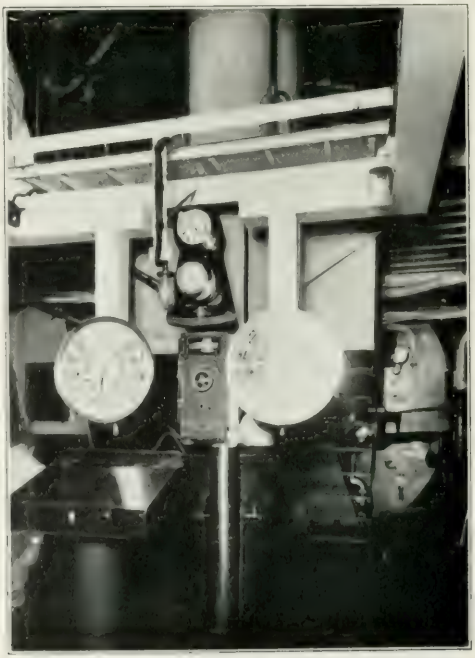
Thrust Block and Centrifugal Governor on L.P. Turbine.



A Weir Lunette type Evaporator in the Engine Room.

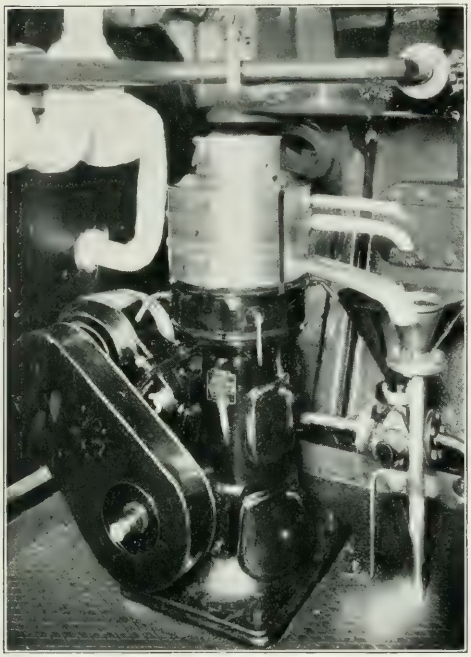


Stokehold Control Telegraphs and Navyphones in the Engine Room.

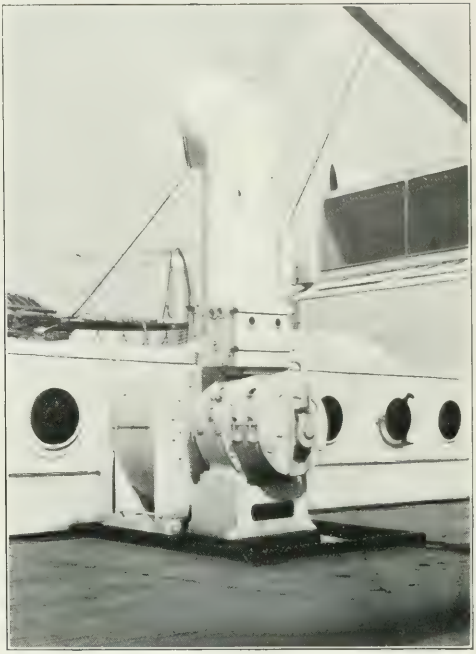


View of the Control Position, showing Telegraphs, Clocks, and Distance Thermometer.

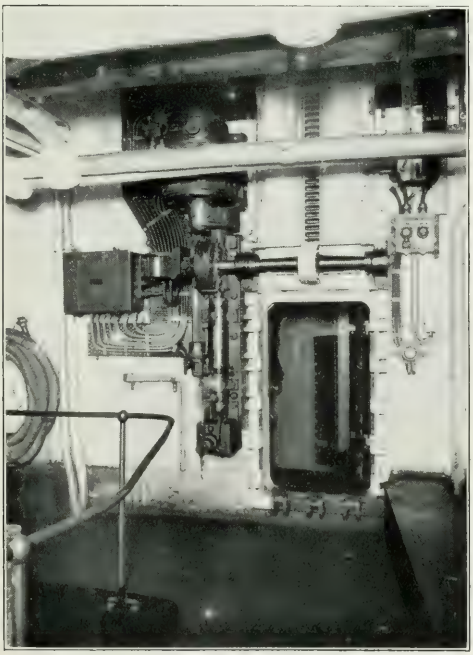
RED STAR LINER "BELGENLAND."



View showing the De Laval Centrifugal Oil Purifier in the Engine Room.

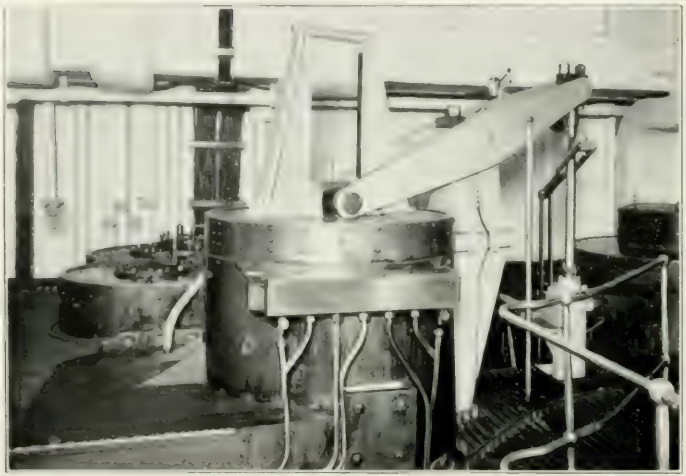


View showing the Exhaust Vent and Fan from the Smoking Room.

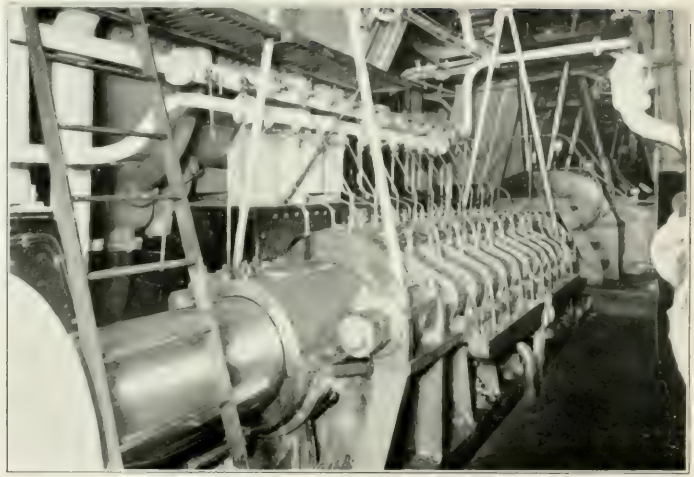


Harland & Wolff Patent Watertight Door leading into Shaft Tunnel.

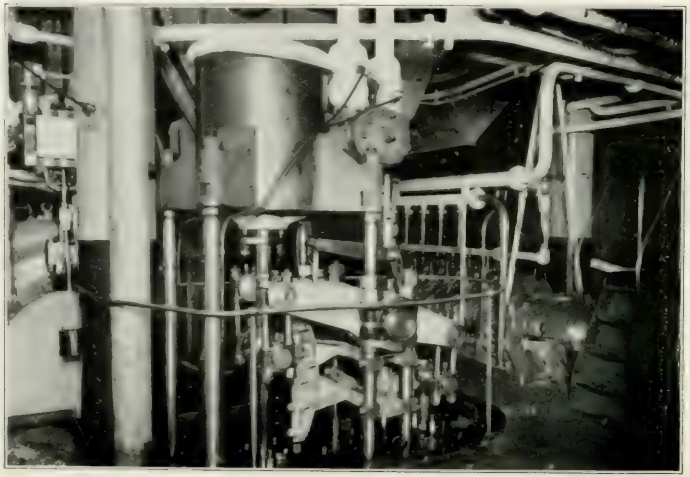
Cylinder Tops of the Port Main Engine, showing H.P. Cylinder Cover Lagging Removed.



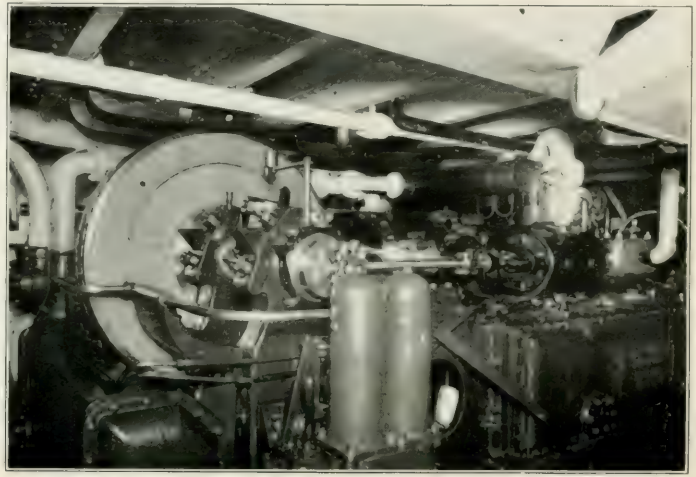
The Port Change-over Valve.



The Starboard Main Engine Thrust Block.



Weir's Wet and Dry Air Pumps in Engine Room.



Refrigerating Machinery on Port Side of Engine Room.

each engine passed from the low-pressure cylinder direct into its respective condenser. The engines driving the two wing shafts are of the four-cylinder triple-expansion reciprocating type, balanced on the Yarrow, Schlick, Tweedy system, the cylinder dimension and the order being as follows :—

Cylinders.	1st L.P.	H.P.	I.P.	2nd L.P.
Diameter	64 in.	35½ in.	56 in.	64 in.
Stroke	5 ft.	5 ft.	5 ft.	5 ft.
Piston rings	Ramsbottom.	Campbell & Banks.	Lockwood & Carlisle.	Ramsbottom.

Aspinall governors are fitted to control the Brown's engines which operate the throttle valves, and emergency gear is arranged to cause the links to move to mid-position in case of excessive speed. The two engines collectively develop an indicated horse-power of 11,230.

The low-pressure turbine, driving the centre shaft, is of the Parsons' impulse reaction type, having six expansions, with left-hand blading driving a right-handed propeller, running at about 185 r.p.m. The following are the chief particulars of the turbine :—

Rotor :—	
Weight	56 tons
Diameter	11 ft.
Dummy piston :—	
Diameter	7 ft. 9 in.
Number of rings	19
Steam inlets :—	
Number	2
Diameter	3 ft. 3 in.
Exhaust outlets :—	
Number	2
Area	787 ft.
Steam pressure from wing engines	12 lbs. absolute (approx.).
Tunnel shaft diameter	14 in.
Oil pressure per sq. in. on bearings	73·5 lb.
Weight of turbine complete, neglecting lifting gear, eduction bends and sluice valves	200 tons

Two large condensers are provided, into which each set of reciprocating engines can exhaust direct if the turbine is at any time not running, as has been mentioned. The two condensers are of the Contraflo type, and were constructed by Harland & Wolff Limited. They are steel units, having a height of 6 ft. 9 in., a breadth at the top of 9 ft. 4 in. and at the bottom of 2 ft. 2 in., and a cooling surface of 8,650 sq. ft. The auxiliary engines and machinery are all independent of the main engines. The thrust blocks for the two wing engines are of the multiple collar adjustable type, each having ten shoes. They are lined with Harland & Wolff white metal ; the propeller shafting is made of Siemens-Martin ingot steel. With regard to the propellers themselves, the two outer ones (which are right and left-handed for the port and starboard engines respectively) are of the built type, having a cast steel boss and manganese bronze blades, whilst the centre propeller is of the solid type, being made of turbadium alloy. The following particulars of the propellers are also interesting :—

Shaft.	No. of Blades.	Diameter.	Mean Uniform Pitch.
	R.H. L.H.		
Wing	3 3	19 ft. 6 in.	26 ft. 6 in.
Centre	Right Handed	13 ft.	—

The manœuvring platform is situated between the two main engines, and we illustrate this on page 208. The view shows the Graham patent navyphones with which communication may be effected with various parts of the ship ; the engine-room telegraphs supplied by J. W. Ray & Co. Ltd. will also be noticed. The distance thermometer, which will be seen between the telegraphs, is of the type made by the Cambridge and Paul Instrument Company Limited ; this registers the temperature of the uptakes to each group of six fires, and the exact state of each uptake may be determined by moving over the variable contact on to its respective stud. The manufacturers of the engine-room telegraphs were also responsible for the special stokehold indicators, which will be seen on the same page.

Boilers.

Steam is supplied by ten double-ended boilers, each having six furnaces of Morison type, withdrawable on the Ashlin principle. The boilers are constructed from Siemens-Martin steel, and have each a mean diameter of 15 ft. 9 in., and a mean length of 20 ft. Each of the furnaces has an effective diameter of 3 ft. 9 in. and a total length of 8 ft. 9⅜ in. The working pressure for the boilers is 215 lb. per sq. in., and there is no superheat. Constructed originally to burn coal fuel, they now burn oil on the White low-pressure system ; there is no special system of draught fitted, and the boilers work under what may be described as an assisted draught, there being eight electrically-driven fans available when necessary.

Oil Fuel System.

The White system of oil fuel burning is fitted. The furnace front fittings, which are shown on page 207, are of the latest design, having a double air-cone with spiral vanes and valves at the outer end of the cone, which permit of a much closer adjustment of the air supply round the burner and through the Venturi ring, than with the earlier designs. A new safety cock is fitted close to the burner, so arranged that the locking pin for the burner yoke cannot be turned until the safety cock which controls the fuel oil supply to the burner is closed. The fuel generally used is Mexican oil having a flash point of 240 deg. and a specific gravity of 0·958 ; the working pressure is usually between 70 and 80 lb. per sq. in., and the diameter of the burner 0·059 in. Part of the starboard forward oil fuel unit is shown on page 206.

There is a pumping and heating unit in each boiler room, and the complete unit is made up of the following components :—

Two Weir's double-acting pressure pumps, either of which deals with oil for one boiler room at full power.
Three oil fuel heaters (one reserve).
One sediment pump.
One Weir's transfer pump.
One suction oil fuel filter.
One discharge oil fuel filter.

The fuel pumps are fitted with filters in duplicate on the suction and delivery sides of the pumps. In the forward boiler room there is a sediment pump arranged to draw from oil gutterways, drain pots in fuel tanks, &c., and to discharge into an inspection tank convenient to the pump, for the purpose of ascertaining whether the pump is drawing oil or water. The danger of fire in the boiler-rooms is guarded against by means of Foamite patent fire-extinguishing units, and we illustrate on page 211 the auxiliary unit in No. 4 boiler-room, which is of the 34-gallon type, working at 100 lb. steam pressure. The main Foamite installation consists of two large tanks placed in the engine room, and there are two similar tanks in the forward pump room ; the contents of these tanks are discharged by air maintained under pressure by a Reavell three-stage compressor. In addition to the storage tanks there are also the 34-gallon Foamite tanks in each stokehold similar to the one we illustrate. These are discharged by direct steam pressure on the liquids.

Auxiliary Machinery.

The *Belgenland's* auxiliary equipment is very complete. The circulating pumps, of which there are four complete units, are situated well aft in the engine-room. They are of the centrifugal type, made by the engine builders, and each is driven by a high-speed reciprocating engine having a cylinder

diameter of 12½ in. and a stroke of 5 in. The diameter of the impeller is 5 ft. 1 in., and the amount of water which can be discharged by the pumps per hour collectively at full power is 5,700 tons. The main feed pumps are of the G. & J. Weir type, having a cylinder diameter of 18½ in. and a pump diameter of 13 in. There are two pairs of these pumps, arranged near the boiler-room bulkhead, one on each side of the ship, and we illustrate the starboard pair on page 206. The main air pumps are of G. & J. Weir's dual type, and there is one situated on either side at the rear end of the low-pressure turbine. Details of the auxiliary condensing apparatus will be found in the table below, together with particulars of most of the horizontal duplex type and we illustrate a portion of it on page 210.

It works on the CO_2 system, and the brine tank and brine circulating pumps are situated above the machine itself; the pumps are by Thom, Lamont & Company, and there are three in all. The system of cold distribution used is the actual circulation of cold brine. The engine of the refrigerating machine has a high-pressure and a low-pressure cylinder, the diameters of which respectively are 12 in. and 23 in., steam being supplied to the engine at a pressure of about 215 lbs. per sq. in.; there are two compressors, each having a diameter of 6 in. and

Details of Auxiliary Machinery.

Function.	Pump, &c.	Type.	Maker.	No. Off.	Remarks.
Main steam unit—	Feed	Direct acting	G. & J. Weir	2 pairs	Cylinder diameter, 18½ in.; Pump do., 13 in.
	Air	Dual	G. & J. Weir	2	At aft end of L.P. turbine.
	Circulating	Centrifugal	Harland & Wolff	4	Total capacity, 5,700 tons per hour.
	Condenser	Contraflo	Harland & Wolff	2	Cooling surface, 8,650 sq. ft.
Auxiliary steam unit—	Feed	Direct acting	G. & J. Weir	1	Cyl. diam., 10½ in.; pump bore, 8 in.
	Air	Monotype	G. & J. Weir	1	Cyl. diam., 14 in.; pump diam., 22 in.; exhausts to auxiliary condenser.
	Circulating	Centrifugal	Harland & Wolff	1	Steam-engine drive, 7 in. × 7 in.; pump sea suction diam., 9 in.
	Condenser		Harland & Wolff	1	On starboard side.
Bilge and ballast pumps—	Ballast	Duplex	Harland & Wolff	3	Suctions on pump and sizes of bore:— Suction Detail. Bore. Sea and tank 9 in. Emergency bilge 8 in. Fore and aft 5 in. Bilge direct 5 in.
	Bilge	Duplex	Harland & Wolff	2	
Fresh water, &c.—	Fresh water	Vertical	G. & J. Weir	2	—
	General service	Duplex	Harland & Wolff	2	Cylinder diameter, 12 in.
	Sanitary	Duplex	Harland & Wolff	2	Pump stroke, 15 in.
Turbine forced lubrication	Oil	Vertical	G. & J. Weir	2	Pump diameter, 7 in.
Main reversing engine	—	—	Brown Bros., Edinburgh	1	Steam cylinder, 17½ in. diameter; water cylinder, 8½ in. diameter.
—	Oil purifier	De Laval centrifugal	Chadburn's	1	825 r.p.m.
Main engines	Turning engine	—	Harland & Wolff	—	Cylinder diameter, 8 in. diameter × 9 in. stroke; piston rod diameter, 1⅝ in.
Turbine lifting gear	Motor	—	Laurence Scott	1	B.H.P., 20; amps., 175; R.P.M., 266.

bilge, ballast and other pumps. The two evaporators are of G. & J. Weir's Lunette type. One of these is shown on page 208. Steam is supplied from a live steam pipe of 2-in. bore, at 215 lb. pressure (*i.e.*, at boiler pressure). The blow-off pipe is led to the ship's side, and has a bore of 2½ in.; the brine pipe has a bore of 1½ in. and is led to the bilge. The complete weight of one evaporator is 2 tons 3 cwt. 1 qr.

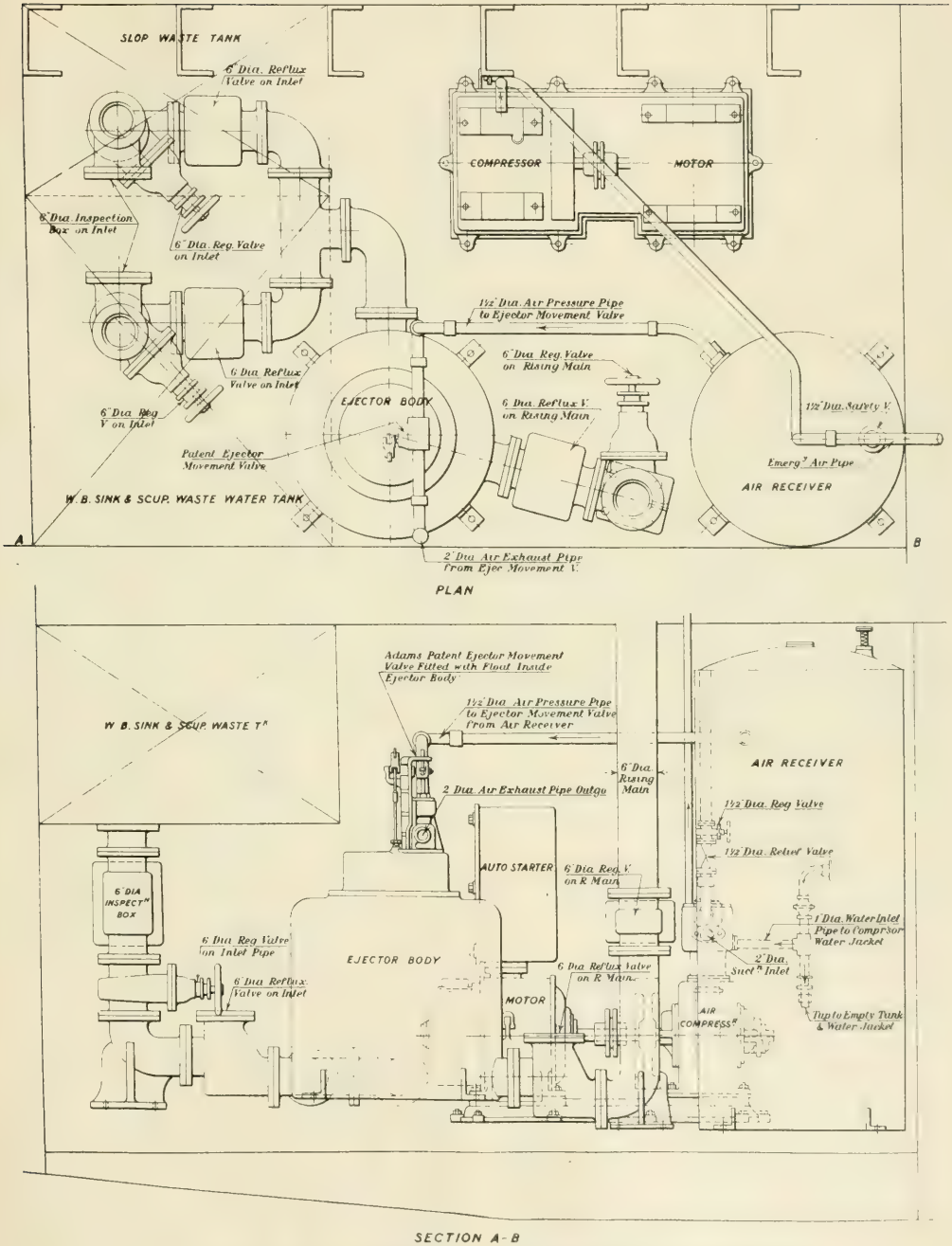
There are two distillers placed in the engine-room recess on the lower deck, starboard side aft. The capacity of each still is 5,000 gallons in 24 hours; the circulating water inlet has a bore of 3½ in., and the outlet has the same bore. The total weight is 8 cwt. 3 qr.

Refrigerating Machinery.

The refrigerating machinery is situated on the starboard side of the engine-room, and consists of a single set supplied by J. & E. Hall, Limited, Dartford. It is one of their No. 11 a stroke of 15 in. The machine refrigerates 12 domestic chambers, two cargo chambers, 20 cold cupboards and 20 ice water fountains.

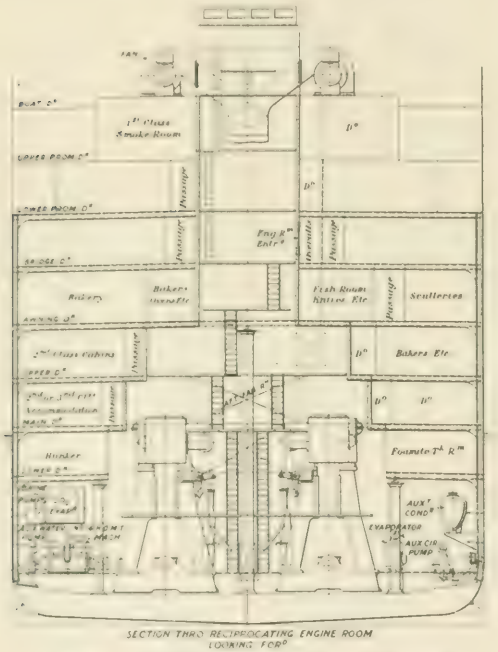
Electric Generating Sets.

These are situated in a special compartment abaft the main engine-room; the compartment is bordered, as it were, on its two sides by the wing shafts, and the centre shaft runs through the compartment. The complete power plant consists of four units, each of which comprises a generator giving 3,000 amps. at 100 volts, *i.e.*, 300 kw., driven by a compound engine developing 440 b.h.p., and running at 330 r.p.m. The steam pressure is 150 lb. per sq. in., and the cylinder diameters are 16 and 25 in. respectively by 12-in. stroke. The generators and engines were supplied by W. H. Allen Sons, & Co. Ltd., Bedford. The main switchboard is arranged above the generators about midway along the compartment on a platform stretching the full width


SLOP WASTE TANK
6" Dia. Reflux Valve on Inlet
COMPRESSOR
MOTOR
6" Dia. Inspection Box on Inlet
6" Dia. Reg. Valve on Inlet
1½" Dia. Air Pressure Pipe to Ejector Movement Valve
6" Dia. Reg. Valve on Rising Main
6" Dia. Reflux Valve on Inlet
EJECTOR BODY
6" Dia. Reg. V. on Inlet
6" Dia. Reflux V. on Rising Main
1½" Dia. Safety V.
Patent Ejector Movement Valve
Emergy Air Pipe
AIR RECEIVER
W.B. SINK & SCUP. WASTE WATER TANK
A
B
2" Dia. Air Exhaust Pipe from Ejec Movement V.
PLAN
Adams Patent Ejector Movement Valve Fitted with Float Inside Ejector Body
W.B. SINK & SCUP. WASTE Tk
1½" Dia. Air Pressure Pipe to Ejector Movement Valve from Air Receiver
AIR RECEIVER
6" Dia. Rising Main
2" Dia. Air Exhaust Pipe Outgo
1½" Dia. Reg. Valve
AUTO STARTER
6" Dia. Reg. V. on R. Main
1½" Dia. Relief Valve
6" DIA INSPECTN BOX
1" Dia. Water Inlet Pipe to Comprsor Water Jacket
6" Dia. Reg. Valve on Inlet Pipe
EJECTOR BODY
2" Dia. Suctn Inlet
6" Dia. Reflux Valve on Inlet
6" Dia. Reflux Valve on R. Main
MOTOR
AIR COMPRESSR
Tap to Empty Tank & Water Jacket
SECTION A-B

The Red Star Liner "Belgenland." Lay-out of Sewage Unit. (*See page* 216.)

of the compartment. In the generator compartment are fitted also the two rotary transformers of the Cumberland Electrolytic System, which is fitted to the main boilers and condensers for the prevention of corrosion. These machines, which were specially made for the Cumberland Engineering Company by Laurence Scott & Co., are shunt wound and designed to run off the ship's power circuit and they give a secondary output of 200 amps at 10 volts d.c. In case the main generating sets should give out through the compartment being flooded, there is an emergency generator situated in a special compartment at the aft end of D deck. The prime mover for this consists of a



T.SS. "Belgenland." Section through the Machinery Space.

four-cylinder Diesel engine running at 400 r.p.m., and the generator is by Laurence Scott having an output of 750 amps. at 100 volts.

Sanitary Arrangements.

In order to obtain a satisfactory means of discharging solid and waste from certain lavatories on the main deck owing to the close proximity of these to the load water line, the discharges from these spaces are led into sludge tanks placed at different positions throughout the machinery spaces; the soil is then drained by means of gravity into ejector bodies in which a float is arranged; this float on rising opens a valve connected to an air reservoir, which, when opened, releases the air and blows the entire contents of the ejector overboard. When the ejector is emptied the float drops and shuts the air valve, the sewage again flows into the ejector, and the above operation is repeated until the soil tanks are emptied. The air for ejecting the sewage overboard from the ejector is stored in an air receiver at a predetermined pressure, and is maintained at this pressure by an air pressure contact switch which is connected between the air reservoir and motor and air compressor. The electro-pneumatic sewage lifting plant has been supplied by Adams Hydraulics Ltd., and we show the lay-out diagrammatically on the preceding page.

THE PROPELLING MACHINERY OF PLEASURE STEAMERS.

A correspondent who signs himself "London River" sends us the following communication bearing on the editorial article which discussed the propelling machinery of paddle steamers in our last issue. His letter reads as follows:—

"I was much interested in your editorial article on p. 161, dealing with the Propelling Machinery of Pleasure Steamers, but I think you are rather 'down' on the paddle wheel. I always understood they were much easier to manœuvre than screws, there being no cutting action, and on the Thames it is a common thing for the vessels (which are fitted with rudders at each end, but have not disconnecting engines) to swing two or three miles before reaching their final destination on the return trip, and complete the journey stern first. I heard a suggestion some years ago of fitting a paddle boat with triple-expansion engines geared on to the paddle shaft, so as to get the advantage of light, quick-running machinery, and I should be glad to know whether Diesel engines would be any less in height and/or weight than steam, and whether there is any objection to a friction drive for the gearing."

Our correspondent in his letter has opened up for discussion a subject at once controversial and full of interest. Paddle-wheel *versus* screw for tugs, for instance, has always been a vexed question, and the Admiralty especially have generally shown a strong inclination for paddle tugs, not without sound reasons, since they contend that when a paddle tug is tied alongside a large ship her outboard paddle is able to exert a far greater leverage for turning than would be possible in the case of a screw tug. Also, of course, the paddle tug has the advantage of ease of manœuvrability with independent paddles. But with regard to actual towing, and pulling powers, which after all is the *raison d'être* of the tug, experiment has shown (both the early *Rattler* experiments and later the 1913 Thornycroft trials) that the screw is infinitely superior. The Diesel engine was applied to paddle-wheel craft (in the shape of Russian tugs) as long ago as 1908, and although in certain cases the paddles were not independent of each other, some very useful work was got out of them; various types of drive and gearing were used. With regard to the pleasure passenger steamer, the Diesel engine has distinct advantages, and there seems to be no reason why it should not be extended to pleasure passenger vessels of the paddle type. But the more obvious procedure, if Diesel engines were adopted, would be for them to drive screws either directly or through reduction gearing, and it should be possible to arrange a design in which weight is saved; tug practice, in the smaller types at any rate, has shown that there is often an all-round saving, and head room should not prove a serious difficulty. There would seem to be possibilities in a suitable friction drive for a light high-speed reciprocating engine drive to paddle-wheels, but it seems unlikely that such a drive would be adopted at the present time.

72 HOURS' FULL POWER RELIABILITY TRIAL. The Parsons Motor Company Limited, Southampton, completed recently an interesting 72 hours' continuous full-power run on one of their marine motors. The engine selected was a standard 28-h.p. marine unit having four cylinders, 4½-in. bore by 6-in. stroke, and running at 850-900 r.p.m., and the test was carried out to demonstrate the suitability of this type of engine for full-power work over extended periods. The test was carried out under the supervision of the Admiralty Overseer, and, being started on a Friday afternoon at 4 o'clock, the engine was run right through the week-end and finally shut down at 4 o'clock on Monday afternoon. It is interesting to note that during the run the counter registered no less than 3,621,000 revolutions, which means that each engine piston travelled a linear distance in its cylinder of about 700 miles.

TEXT-BOOK ERRORS IN MARINE BOILER WORK.

Some Examples of Incorrect Diagrams and Drawings.

The author has recently spent much time with marine engineers, and, being a boilermaker, he has often been called upon to explain the construction and repairing of marine boilers to young engineers studying for their examinations.

Certain errors have been noticed in text-books dealing with marine boilers, and an effort will be made here to point out several of these in the hope that something will be attempted by those concerned towards rectifying them, for there are many young engineers who have had no opportunity of seeing boilers under construction who are given false conceptions of the work owing to mistakes in the books issued for their guidance.

Much that is good can be learnt from technical books and improved on by practical knowledge of the work, but unless the book is strictly correct in detail, a false conception will be gathered, and thus many an otherwise clever young engineer may fail to satisfy his examiners and jeopardise his chances of promotion.

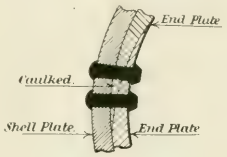


Fig. 1.—Text-Book Method of Scarphing Boiler End Joint.

It is well known among engineers that examinations have, during the last two or three years, become more difficult, and also that more attention is being paid to questions on boilers than was formerly the case, yet the author would suggest that if examining boards included a boilermaker or even an engineer or draughtsman who had made a close study of marine boilers, fully 95 per cent. of the candidates studying from some of our most popular text-books would fail to satisfy the examiners.

Of course it is not the author's intention to criticise text-books in general, but while many good points can be seen in the diagrams and printed matter dealing with boiler construction, the fact remains that they are very often mistaken in detail. In this article an endeavour will be made to point out a few examples of errors found in marine engineering books. It will readily be seen by any draughtsman or boilermaker that these are palpable errors such as are never made in design or construction.

Perhaps some reader may have a book containing such errors as are here quoted; if so, the best course to adopt is to get expert advice from someone well versed in designing or boiler construction before copying some of the examples which are contained in well-known manuals at present being sold. Take, for instance, Fig. 1, which is an exact reproduction of a sketch showing the method of scarphing a boiler end joint.

To the average engineer this may appear quite correct, yet at first glance it is entirely wrong from the point of view of a boilermaker, due to the fact that a double scarph is shown, and such a thing as a double scarph on an end joint is unknown in the design or construction of marine boilers.

The note accompanying the sketch is to this effect: "Special notice should be taken of the fact that the two end plates are scarphed to reduce these plates to the thickness of one where overlapped, and that extra heavy caulking is required to make the joint watertight."

Looked at from any point of view the double scarph, as illustrated in Fig. 1, is not only unnecessary, inasmuch as where three plates overlap in that manner a level bearing is obtained by scarphing that in the centre only. This plate is thinned so as to follow a true line with the radius of the shell, but the inner plate of the three is left fully thick and set back to the shape of the scarph, thus forming a metal-to-metal joint with all three as seen in Fig. 2.

Fig. 1 is contrary to Board of Trade, Lloyd's, or any other rules, and if applied to marine boiler end joints would be found detrimental to good results for each and all of the following reasons:—(1) It would not stand the strain of hydraulic riveting, which is always asked for (where practicable), but would fracture at the landing on the thin end of the inner plate when the tremendous pressure of a modern hydraulic riveting machine was put on the rivets, owing to the fact that the plate was not thick enough to allow of the rivet head and neck being forced into the hole. (2) Even if the joint were hand riveted, the inner scarph would crack between the rivets and at the landing instead of putting a shearing stress on the rivets when the boiler was under steam or water pressure. (3) It would be impossible to caulk such a joint inside owing to the fact that the plate, being reduced to the extent shown, would give no edge on which to allow a caulking tool to be worked, but would crumple up over about 40 per cent. of the length of the scarph and also over the entire depth of the flange from toe to heel as there would be no backing to carry the weight of the blow when caulking. (4) When subjected to corrosive action the inner scarph would waste away and probably fall from the rivets after the boiler had been only a short time in use. Were it possible to obtain good results from such a joint, the present Board of Trade rules applying to plate strength, rivet strength, pitch, diameter, and shearing would be useless. These rules, however, must still apply, as also must the formulæ based on

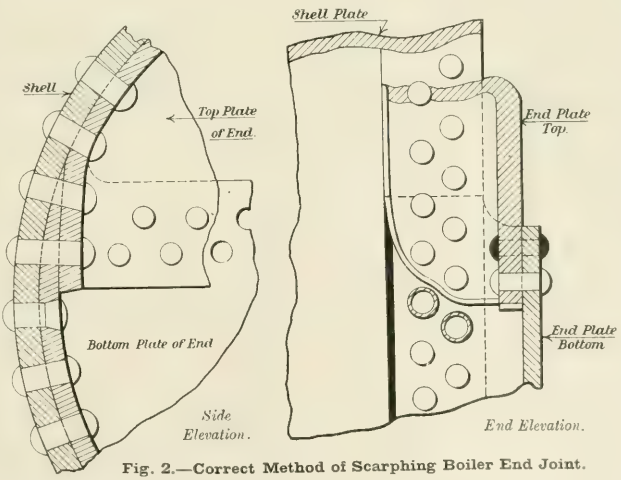


Fig. 2.—Correct Method of Scarphing Boiler End Joint.

a rivet diameter equal to the thickest plate in such a joint, but by no means on the thickness of two plates reduced to that of one.

The only known instances where two plates in a joint are scarphed, occur where the butt straps are overlapped by the end flange (or by the shell plates in circumferential seams of double-ended boilers or two-strake shells). Even where this exists in practice the thickness of the plate is not reduced to a great extent and sufficient edge is left to allow for caulking. In many shops the inner scarph in this case is often dispensed with, for where a good hydraulic riveter is available, the flange is simply set to the shape of the butt-strap scarph, thus the work is performed much quicker than a scarph could be cut and the plate retains its original thickness and strength. In the note accompanying the sketch, shown in Fig. 1, we find the words "Extra heavy caulking is required to make the joint watertight," which piece of advice makes the matter appear even more ridiculous, for having described a joint on which less caulking could be done than on any plain seam, the advice is superfluous. Fig. 2 is an illustration of a boiler end joint and it will be noticed that this has none of the defects pointed out in Fig. 1.

The inner plate of the boiler end in Fig. 2 is set back over the length of the scarph and the rivets are graduated in length according to the various thicknesses of plate at the holes in the three-ply joint. As only the outer surface of the flange has a bearing on the shell plate, the outer flange only requires scarphing, so it is merely common sense to say that no practical man will weaken his plating or spoil a good job in any way by cutting or drawing out a scarph where it is not required. The inner flange being left at its original thickness is proof against any of the objections pointed out in Fig. 1. First, it is thick enough and strong enough at the landing to stand any type of riveting. Secondly, it would not be split when under steam or water pressure, but would put a shearing stress on the rivets. Thirdly, having a full edge, it will take a caulking tool of any description with equal effect on all parts requiring caulking.

It will also resist the ravages of corrosion for any reasonable number of years, whereas the inner scarph in Fig. 1 could not be guaranteed to do so, even if the other objections were overlooked.

An end elevation of the joint with the cross or horizontal seam in section is shown to the right in Fig. 2, and all covered parts of the scarph and shell plate are denoted by dotted lines. Two countersunk holes are shown next to the joint, as in most shops claiming good workmanship it is the practice to fit here countersunk rivets, thus allowing a caulking tool to pass over when caulking the joint, one such rivet being seen in section in the left-hand view. This is not a standing rule, but is to be preferred to caulking the joint before filling these two holes, as is done in some shops.

In the manual containing the original of Fig. 1 there are, but a few pages away, photographs of a boiler front and combustion chambers, easily recognised by the surroundings and contract number as those for No. 20 boiler of the Cunarder *Mauretania*, and photographed in the boiler shop of the Wallsend Slipway & Engineering Company during the author's apprenticeship.

One has just to glance at these photographs almost side by side with Fig. 1 to find that the methods illustrated in Fig. 2 have been adopted on all four corner joints in the boiler ends, these ends being in three pieces, hence two horizontal seams and four such joints as we are discussing.

In Fig. 3 we have the joint shown in section through both the horizontal seam and through shell plate and flange, the section through the flange at A and B being taken nearly on the tip of the scarph. It will be noticed that from the edge of the shell A to the toe of the flange B the scarph is of equal thickness, for in scarphing by drawing out or by any machine or hand tools, that part of the scarph which is overlapped is tapered only on the radius of the circumferential seam. The tapered section round the heel of the flange—as seen hand riveted in the sketch—is due to the fact that in most cases the scarphing is done before the plates are flanged unless cut in place by hand or pneumatic tools, and in making it broad enough for the overlap to be a perfect joint it naturally follows that in flanging, the tapered part assumes the shape seen here.

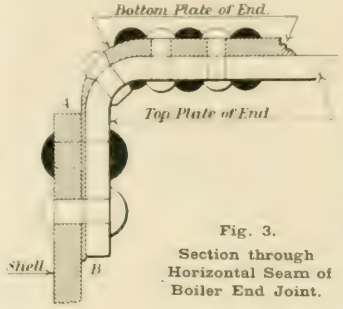


Fig. 3.
Section through Horizontal Seam of Boiler End Joint.

In Fig. 4 from the same text-book as Fig. 1 we have a different type of scarph to any yet seen in the author's experience, for here two glaring mistakes are made, inasmuch as the tapering is carried below the landing edge of the shell plate, thus making an imperfect scarph such as would not give a metal-to-metal joint without deforming any plate that overlapped it. Neither is the scarph carried up to beyond the edge of the inner plate of the joint, thus weakening it, and only applying single riveting as shown, whereas the joint is double-riveted. Further failing to carry the scarph through the entire width of the overlap makes it impossible to caulk the joint as it ought to be caulked and as it would be at B in Fig. 3.

In the same book as that from which the examples in the first portion of this article were taken, we have Fig. 5, and by the sketch and accompanying notes we are asked to believe that a double-scarphed joint applies where the wrapper-plate, tube-plate and furnace meet as shown at 1 and 2. Such a joint would be even worse in practice than those already seen in Figs. 1 and 4. Not only would it embody most of the faults above pointed out, but in addition we must bear in mind that countersunk-headed rivets would here be used. Let us suppose the tube-plate is $\frac{7}{8}$ in. thick and the rivets $\frac{3}{4}$ in. diam. The usual rule for countersinking is: depth = three-fifths thickness of plate with an angle of 45 deg. How are we to apply the rule on a scarph which tapers from $\frac{7}{8}$ in. to nothing? Certainly a knotty problem, for the depth of countersink required for the head of a $\frac{3}{4}$-in. rivet would go well into the second plate in the joint. One more important point overlooked in the book is the effect of fire on such a joint. There would be little material remaining if an uncovered scarph were exposed to the flame during a few days' steaming, but apart from that objection we can repeat without fear of contradiction all that has already been said here of such joints in any position on boiler construction. Such have been seen in woodwork (and the author himself has made double-scarphed joints on safety and time fuses while serving in the Royal Engineers), but never

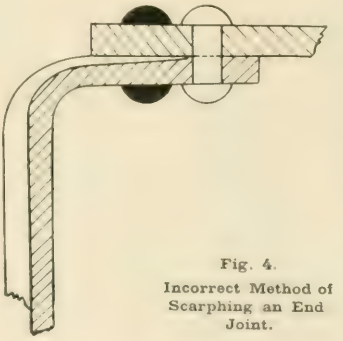


Fig. 4.
Incorrect Method of Scarphing an End Joint.

in boiler construction, where they would be as ridiculous as they are unnecessary.

In Fig. 6 is shown the correct method adopted in making these joints where a non-withdrawable furnace is in use. Note that the furnace is not scarphed but is thick, making a caulking edge and covering the scarph. Where the Ashlin type of furnace is used the furnace, instead of the tube-plate, is scarphed, for being withdrawable it is on the water side of the tube-plate instead of the fire side, thus it is overlapped at the position where it meets the tube-plate and wrapper-plate by both of these, and it therefore forms the scarph. It is possible that when the book in question was being written some confusion may have arisen through seeing various types of furnaces; but the fact remains that the error has been made and the book containing it sold broadcast to many who accept it as describing genuine workshop practice. It is the author's endeavour here to correct any misconception thus derived, for it is highly improbable that any engineer will be granted a certificate should he sketch or describe such joints as we see in Figs. 1, 4 or 5.

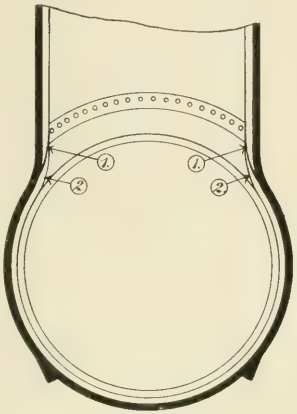


Fig. 5.—Double Scarphed Joint.
1. **Scarph of Tube-Plate.**
2. **Scarph of Furnace-Plate.**

Take next the definition of caulking as given in most of the books now on sale, *i.e.*, caulking is the process by which seams are made watertight after riveting, a blunt chisel about $\frac{1}{4}$ in. broad being used to burr up the edge of the plate. The foregoing is typical of the definition contained in books which the author has seen recently. "A blunt chisel" has nothing whatever in common with a caulking tool, being a blunt chisel and nothing more; but a caulking tool is a tool specially set and ground according to the job for which it is required. There is no fixed breadth such as $\frac{1}{4}$ in., for one may as well attempt caulking with a toothpick as use such a tool in the hope of making a boiler seam watertight for any length of time. Caulking tools vary in breadth according to the thickness of the plate, the test pressure, and also according to the workmanship put into the plating and riveting of the work. With modern high pressures the $\frac{1}{4}$-in. tool is seldom used in marine boilers except inside the impression left by a broader tool, thus giving what is termed "double caulking."

Caulking tools usually vary from $\frac{5}{16}$ in. up to $\frac{1}{2}$ in. in breadth on ordinary plain seams, with pneumatic or hand caulking, while in the case of shell butts and scarph joints such as Fig. 2, and on the lower parts of circumferential seams, where faulty circulation often causes leakage, the best of builders have these parts first hardened up with tools $\frac{5}{8}$ in. or $\frac{3}{4}$ in. broad before applying the smaller tools. On Admiralty work, where no impression is allowed on outer seams, the tool must be as broad as the plate is thick, and is commonly termed a staving tool.

The tools also vary in shape, length, breadth, &c., to meet the circumstances of the job, and are too numerous and various to be detailed here. It should be noticed that in caulking, the plate edge is not simply burred up, but the material is first gathered towards the seam by the sharper edge of the tool which has been specially bevelled for this process, which is known as "splitting in," after which the tool is reversed and the long side of the bevel drives the material down into any hole remaining in the seam. This latter process is commonly termed setting, and is the finishing process.

Fig. 7 gives an incorrect sketch of a seam after caulking, as illustrated in another well-known manual. Notice the impression supposed to have been left by a caulking tool. It is a sheer impossibility to make a seam tight with a tool that would leave a similar impression, for it would be shaped so as to strike out from the inner plate and would thus be nearly impossible to hold while being hammered, and further, it would have a tendency to drive the material away from the seam—exactly the reverse process from caulking. Probably a "blunt chisel" might make such a mark, but certainly no caulking tool.

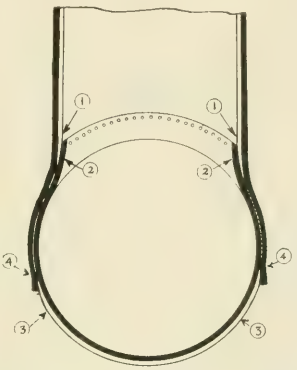


Fig. 6.—Correct Method of Making Furnace Joints.
1. **Tube-Plate Scarphed.**
2. **Furnace-Plate not Scarphed.**
3. **Bottom-Plate Scarphed.**
4. **Wrapper-Plate not Scarphed.**

In Fig. 8 we give an example of a boiler end seam in section through the shell plate and end flange (hand riveted) showing caulking as commonly practised in most shops. Note the shape of the impressions as compared with those in Fig. 7. The outside tool (that on the reader's right) is as a rule $\frac{1}{2}$ in. or not less than $\frac{3}{8}$ in. breadth, and on the inside seam it is usual to employ two tools, leaving the double impression seen in the sketch. These are usually, first tool $\frac{3}{8}$ in., and the second $\frac{5}{16}$ in. or $\frac{1}{2}$ in. No caulking tool should go deeper than $\frac{1}{16}$ in. into a plate edge unless the seam is badly fitted and badly riveted, in which case broader tools should be used to prevent lifting, *i.e.*, wedging the landing up, although this rarely occurs with plate and tools properly bevelled. To anyone with a knowledge of the work anything similar to Fig. 7 is unthinkable as an illustration of caulking.

Finally, we will deal with what will appear to any mechanic to be the most amazing blunder of any pointed out in these articles, and this, too, is culled from the same book as the examples of scarphs already criticised. In dealing with screwed stays the following note of explanation is given: "These stays are screwed through the plates to be supported and are secured by two nuts in all, one outside the boiler and the other inside the combustion chamber, the stay ends being riveted over the nuts."

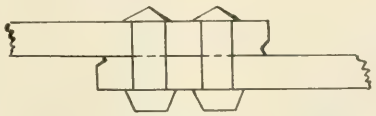


Fig. 7.—Incorrect Diagram of Seam after Caulking.

Accompanying this information there is a detailed sketch which the author does not think it necessary to reproduce; nor is it necessary to add a sketch of his own to illustrate his point, for no sketch is required to point out the blunder contained in such a statement as "the stay ends being riveted over the nuts." The correct statement would be as follows: "These stays are screwed through the plates to be supported, then each

plate is caulked round the end of the stays, after which nuts are fitted to the stay ends thus giving additional support to the plates and minimising the risk of springing the caulking or stripping the threads." We all know there are riveted stays and also nutted stays, but we have yet to see a stay nutted and riveted at the same end, for no craftsman is so unsophisticated as to caulk a stay, nut it and rivet the end over the nut.

Anyone who has had to deal with marine boilers will surely know that no caulking is guaranteed to remain watertight for ever, therefore stays have from time to time to be recaulked, and to caulk a stay it is absolutely necessary that the nut must be removed, unless the people who advocate nutted and riveted stays can suggest another procedure. There are only two ways of dealing with such stays in the event of these leaking or the nuts being burnt. Either to remove the stay and fit a new one or to weld the nut solid to the plate, and either would be a big and expensive job compared with a stay either nutted or riveted.

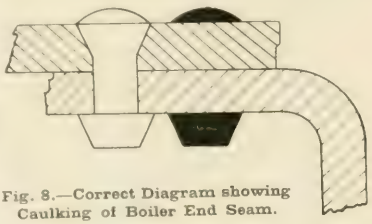


Fig. 8.—Correct Diagram showing Caulking of Boiler End Seam.

Some time ago a company with whom the author was associated had two boilers which when steamed gave a good deal of trouble through leaky chamber stays, these having been caulked with tools $\frac{1}{4}$ in. and in some cases only $\frac{1}{8}$ in. broad. The trial trip was delayed considerably, for the fires had to be drawn and no less than 285 stays recaulked before the job was at all satisfactory. This involved removing the nut in each case. It is obvious that the delay would have been rather lengthy had these stays been riveted over the nuts. If a stay is to be nutted, it is simply nutted and is finished, for besides being detrimental to the future removal of the nut, any hammering would have a tendency to spring the caulking.

When a stay is to be riveted, it is cut much shorter than it it is to be nutted, just sufficient material protruding through each plate to allow for caulking. After being caulked, the stay is riveted over, and then caulked with an ordinary rivet set to finish off, giving it the appearance of a finished rivet.

Should such a stay leak, it is simply caulked in the same manner as a rivet is caulked. Riveted stays are not much in vogue for marine work, except when asked for, but are found where the edge of a butt strap or any such obstruction prevents the easy fitting of a nut. Sometimes they are seen where boilers are close together when shipped, and also in way of boiler seats where these come near side stays.

In conclusion, it may be mentioned that in writing this article the author has made every endeavour to illustrate the errors quoted and hopes it will go far to correct any false conception of the jobs dealt with. In many of the text-books there are numerous deviations from the practical side of the work, and just a few are quoted here. On such a subject as marine boilers no dabbling is advisable, but correct advice should always be sought. Any draughtsman, boilermaker or certificated engineer worth his salt should be qualified to say when a job is detailed correctly.

The young engineer is urged to seek such advice and follow it when studying, for should he in the course of an examination give replies based on some of the errors shown here, it is scarcely probable that he will satisfy his examiners, and in that event the fact that the information appeared in any manual, however expensive, will not avail much for the unfortunate student. Boiler design and construction is no mean subject and it is not to be dabbled in. It should be explained true to workshop methods.

M. ARKLEY.

A NEW SPANISH MOTOR SHIP.

The *Arantza Mendi*, built by the Cia. Euskalduna Construccion y Reparacion de Buques and engined by John G. Kincaid & Co. Ltd., Greenock, for the Cia. Naviere Sota y Aznar, Bilbao, carried out preliminary trials at Bilbao on April 23 and her official trial trip on April 25.

Great interest has been taken in this vessel. She is the first large Spanish steamer to be fitted with the Burmeister & Wain type of engine illustrated on page 230.

The vessel has characteristics and dimensions as follows :—

Dimensions—	
Length b.p.	345 ft.
Length o.a.	360 ft.
Breadth	48 ft. 6 in.
Draught (mean, loaded)	22 ft. 10 in.
Depth moulded	24 ft.
Tonnage—	
Gross registered tonnage	3,500
Deadweight	6,000 tons.
Displacement (load)	8,603 tons.
Displacement (light)	2,603 tons.
Decks	3
Rate	7
Length of engine room	40 ft. 4 in.
Weight of main engine	220 tons.
Shaft horse power of main engine	1,800
Daily fuel consumption	34 tons (loaded at sea)
Bore and stroke of main engine	29·134 in. by 59·055 in.

The vessel's propelling machinery has been constructed on the Burmeister & Wain-Harland & Wolff principle of the single-screw, long stroke crosshead type, having six cylinders each 29·134-in. bore and 59·055-in. stroke, capable of developing 1,800 b.h.p. when running at 90 r.p.m. All the auxiliaries are electrically driven, and in the engine room on the port side are installed three 65-kw. two-cylinder Diesel engine sets. There are two electrically-driven jacket-cooling centrifugal pumps, two for piston cooling, two lubricating oil pumps, one auxiliary air compressor, a fuel transfer pump and the usual ballast, bilge and general service pumps. On deck are ten electrically-operated winches. The steering gear is also electrically controlled. A donkey boiler has been installed for heating purposes, and there is an evaporator together with a small steam-driven air compressor for emergency purposes.

The *Arantza Mendi* is a single-screw ship and her engine is a large, slow-running Diesel. She is a well-proportioned vessel with forecastle, bridge and a large deckhouse aft, and she has a large funnel amidships, which gives her a well-balanced profile.

The order for the vessel's machinery was placed with John G. Kincaid & Co. Ltd., Greenock, who some short time ago took took out the licence for the construction of the Burmeister & Wain-Harland & Wolff type of Diesel engine, and since the time for delivery was very limited they received the assistance of their licensors, Harland & Wolff Ltd., in the carrying out of the contract. The machinery parts were shipped from Greenock and Glasgow to Spain and fitted on board the *Arantza Mendi*, and on the trial trip everything worked well and gave satisfaction to the owners; the speed attained was about 13·3 knots. After completion of the trial trip the vessel left for Philadelphia, and her future progress will no doubt be followed with great interest by Spanish owners.

THERMAL AND MECHANICAL EFFICIENCIES IN STEAM AND MOTOR SHIPS—II.

The Economy and Heat Losses in large Oil Engines—The prospects of Improvement—The scope for the Internal-Combustion Turbine.

(*Continued from page* 185.)

Two detailed sets of oil engine trial results are available, viz., Captain Sankey's report on the Scott-Still engine tests and Professor Watkinson's results from the Camellaird-Fullagar engine. In the latter, the heat balance was given in detail; for the former, it can be calculated from the ample trial data. The results are extremely typical and are of great value in indicating the distribution of heat losses.

(*a*) CAMELLAIRD–FULLAGAR ENGINE.

Tested 31–3–1922.

Two opposed cylinders, 18½ in. by 25 in. (4 pistons).

B.H.P.	1,126.	
R.P.M.	121·1.	
Oil per b.h.p.	0·425 lb. per hour.	
Blast pressure	950 lb.	
Scavenge pressure	1·9 lb.	
Exhaust temperature	430° F.	
Heat in indicated work		44·4
Heat carried away in cylinder cooling water		21·0
Heat carried away in piston cooling oil		3·6
Heat lost in exhaust		24·8
Heat lost by radiation, &c.		6·2
		100·0
Heat in indicated work		44·4
Heat converted into brake work	32·7	
Heat expended in driving scavenge pump	2·9	
Heat expended in driving air compressor	3·54	
Heat in friction of engine	5·26	

(*b*) SCOTT-STILL ENGINE.

Tested 5 and 6–10–1921.

Single oil cylinder, 22 in. by 36 in. (with auxiliary steam cylinder for trial).

Total I.H.P.	394·7	
Total B.H.P.	345*	
Net B.H.P.	327·3*	
R.P.M.	125	
Oil per net b.h.p. solid injection	0·387	
Scavenge pressure	2·25 lb.	
Heat converted into brake work		36·4
Oil	32·2	
Steam	4·2	
Mechanical loss		7·53
Oil	4·96	
Steam	0·6	
Scavenge	1·97	
Heat supplied by regenerater		56·07
		100·0
Heat supplied to regenerater		56·07
Lost in exhaust	11·15	
Converted into steam	44·92	

In this trial engine, of the 44·92 per cent. converted into steam, 4·8 represented the steam indicated horse-power developed and 40·12 was rejected into the condenser. This is roughly 89·5 per cent. of the heat in the steam which was thrown away in this case. This amount would have been greatly reduced with improved arrangements for the expansion of the steam, but of the heat supplied to the regenerater, even at best, some 60 to 70 per cent. will be rejected to the condenser.

The gases left the engine at 678° F., and the feed heater at 164 deg., so that the efficiency of the regenerater appears to have been just 80 per cent.

Owing to the scavenge pump horse-power having been supplied from an external source and not by the regenerater as is obviously possible, Captain Sankey deducted this from the brake horse-power actually developed, but stated clearly that with the exhaust steam from the engine used to drive this pump, a brake efficiency of 39·9 per cent. is obtainable.

* Scavenge h.p. deducted.

(*c*) THE DOXFORD ENGINE.

Tests of this engine have not been published in detail. Mr. Keller gave the following figures before the North East Coast Institute of Engineers and Shipbuilders in January, 1922:—

Four-cylinder opposed piston engine (8 pistons), 580 mm. by 1,160 mm.

I.H.P.	3,146.
S.H.P.	2,540.
R.P.M.	76·2.
Lbs. oil per s.h.p. per hour	0·417.

These actual, and the builder's promised results in their communications to the North East Coast Institute, are given in Table III. It is noticeable that Messrs. Doxford, Scott, and Cammell-Laird have offered consumptions which are, respectively, 2½, 5, and 12 per cent. better than the test bed results show. The average of all the brake thermal efficiencies listed is 35·0 per cent. The interesting and significant features in this table are the fluctuations in the thermal and mechanical losses, the Scott-Still, Doxford and Vickers engines with solid injection showing to considerable advantage as regards the latter. From the results of the Sankey trials, the 38·3 per cent. brake thermal efficiency of the Scott-Still engine should be readily obtainable, and, in fact, had the scavenge pump on test been driven by regenerated steam, would have then been reached. Burmeister & Wain Limited, whose engine is conspicuous by its absence, claim a consumption of but 0·365 lb. per brake horse-power per hour for the main engines only of the *Afrika*, a twin-screw ship of 4,500 i.h.p. at 115 revolutions. With oil at 18,000 B.Th.U. lower value this is equivalent to 38·8 per cent. brake efficiency. Scott's figure of 0·37 is for all purposes. In Table III, however, the losses and efficiency are calculated as if 0·37 lb. per brake horse-power was for main engines only. The fact that builders were given latitude in choice of revolutions and that these ranged from 85 to 112 to some extent vitiated the comparative nature of the different proposals, but as the problem set was, in fact, the propulsion of a 370-ft. 9,220-ton displacement vessel at 10¾ knots trial speed, presumably each builder chose his most economical engine. It is interesting to compare the results on the basis of the required heat energy supply for engines only and for all purposes.

The North East Coast proposals reduced to a common basis.

The seven shots at the North East Coast target require a little calibrating before their precise degree of accuracy in comparison with one another is determinable, and the proposals all require to be reduced to a common basis before their relative merits, as far as fuel economy is concerned, can be assessed. When this is done, there is seen to be a considerable degree of divergence between the various estimates; in fact, it is so great as to call for very careful consideration. As the horse-powers and the consumptions per horse-power, and also the prices and calorific values of the fuels required, varied considerably, we have tabulated in Table V the total quantities of fuel required and the total quantity of thermal units corresponding thereto, all of which figures are clearly given by the different builders. For main engines only, there is a difference in heat energy required of no less than 18 per

MARINE OIL ENGINE EFFICIENCIES AND LOSSES.

TABLE III.—*Data supplied to North East Coast Institute, and data calculated therefrom.*

Type of Engine.	Calorific Value of Fuel. Gross	Net.	Lbs. per B.H.P. per hour.	Mechanical Efficiency.	Brake Thermal Efficiency.	Thermal Loss.	Mechanical Loss.	Total Energy Loss.
Armstrong-Sulzer, 2-stroke		18,000	·386	74·0	36·6	50·5	12·9	63·4
[illegible] opposed pistons, 2-stroke	19,000	17,800*	·39	73·0	36·7	49·7	13·6	63·3
Doxford opposed pistons, 2-stroke	19,200	18,000	·40	82·0	35·4	56·8	7·8	64·6
Neptune, 2-stroke		18,500	·43	73·0	32·0	56·2	11·8	68·0
Scott-Still, 2-stroke		18,000	·37	88·0	38·3	56·5	5·2	61·7
Vickers, 4-stroke		18,000	·42	78·0	33·7	56·8	9·5	66·3
Werkspoor, 4-stroke		17,800	·41	73·0	34·9	52·5	12·6	65·1

* Estimated all purposes.

TABLE IV.—*Test data from Naval Engines.*

Camellaird-Fullagar		18,260	·425	73·8	32·7	55·6	11·7	67·3
Doxford	18,860	17,700*	·417	80·6	34·5	57·2	8·3	65·5
Scott-Still		18,053	·387	87·4	36·4	58·1	5·5	63·6

* Estimated.

TABLE V.—*A comparison of the North East Coast I.C. Engine proposals.*

Builder.	A.-S.	C.-F.	D.	N.	S. S.	V.	W.
B.H.P. (main engine)	1,840	1,828	1,850	1,840	1,870	1,760	1,848
Lbs. oil per b.p. per hour—Main engines	0·386	0·39	0·40	0·43	—	0·42	0·41
All purposes	0·512	0·41	0·41	0·45	0·37	0·43	0·45
Total lbs. of oil per hour—Main engines	710	713	740	791	—	739	758
All purposes	942	749	758	828	692	757	831
Total B.T.U. required per hour— Main engines	12,780,000	12,691,400	13,320,000	14,683,500	—	13,302,000	13,492,000
All purposes	16,956,000	13,332,200	13,654,000	15,318,000	12,456,000	13,636,000	14,791,800
Heat transmitted to shaft*	4,977,000	4,657,740	4,715,280	4,682,670	4,770,648	4,482,774	4,708,708
Heat absorbed in friction*	1,648,600	1,726,030	1,028,960	1,726,750	647,712	1,263,690	1,699,992
Heat rejected in cooling water and exhaust gases*	6,453,900	6,307,630	7,565,760	8,224,030	7,037,640	7,555,536	7,083,300
Fuel consumption per ton-mile for 6,570 d.w., 10½ kts., and [illegible] per day	·00459	·00361	·00369	·00415	·00337	·00369	·004

* See Table III for percentages.

cent. between least and most; for all purposes, this gap is as much as 36 per cent.

In order of fuel economy, both actual and relative, the various types stand as follows, and, on the assumption that the effective horse-power required for the hull is 1,000, the thermo-propulsion efficiency is added.

TABLE VI.

Type.	Actual Energy in B.Th.U. ÷ 10,000.	Relative Energy required.	Therms propulsive efficiency.
Scott-Still	1·246	1·000	20·48
Camellaird-Fullagar	1·333	1·070	19·11
Vickers	1·364	1·095	18·68
Doxford	1·365	1·096	18·66
Werkspoor	1·479	1·186	17·24
Neptune	1·532	1·231	16·62
Armstrong-Sulzer	1·696	1·362	15·00

Oil Engine Efficiencies.

This analysis of the thermal and mechanical distribution of the fuel energy is especially interesting, as it enables some extremely useful deductions to be made in regard to the various types of engine. In arranging for the collection of this data, both the North East Coast Institute and the various firms which provided it deserve both the congratulations and the thanks of all those who are thus, for the first time, enabled to study sound comparative data at their leisure. To render the data rather more complete, the engine dimensions are given in Table VII.

TABLE VII.

Type of Engine.	Armstrong-Sulzer.	Camellaird-Fullagar.	Doxford.	Neptune.	Scott-Still.	Vickers.	Werkspoor.
No. of cylinders	4	4	3	4	4	6	6
Diameter and stroke inches	26·77 · 47·24	24×36×2	21·5×42·5×2	26 · 52	25 · 45	28×45	29·9×55·12
Total cylinder volume cub. ft.	61·0	75·4	52·4	64	51	96	134·4
Revolutions per min.	92	85	96	90	106	112	95
Piston speed ft. per min.	725	510×2	680 · 2	780	795	840	895
Cylinder volume swept per min. cub. ft.	5,610	6,400	5,025	5,760	5,400	10,740	12,780
Type of injection	Air blast	Air blast	Solid	Air blast	Solid	Solid	Air blast
Nature of scavenge	Off engine	Off engine	Off engine	Off engine	Exhaust steam turbo-blower	Piston	Piston

Oil Engine Dimensions and Data.

Deductions from Comparison.

The study of losses and how they can be obviated is always a profitable investigation. It is really extraordinary that seven firms should forward proposals with powers and consumptions per unit horse-power, varying as shown in Table V, and yet that, between the highest and the lowest, the difference in heat energy transmitted to the shaft should only work out to less than 17 lb. of oil per hour. On the mean quantity of fuel required per hour for the main engines, this is only 2·25 per cent. For the heat energy needed for overcoming friction of engine and for driving scavenge pumps and air compressors, the difference is very marked, the low figure in the case of the Scott-Still engine being specially noticeable. That it is based on experience is seen from Table IV. Scavenging pumps, as a rule, require about 3 per cent. of the total heat, and the air blast compressors about 3·5 to 4·5 per cent. There is an appreciable saving here in the Still engine. The quantity of heat lost in cooling and in the exhaust gases is always a formidable amount, and averages, in the seven cases, just over 54 per cent., thus leaving 46 per cent. out of which first the initial friction of the engine and then the pumps and compressors must be driven (unless, of course, these are included in the "all purposes" fuel).

This inevitable loss of rejected heat can only be recovered at considerable expense, and if, as in the Still engine, it is used to regenerate steam, it must always be remembered—as was shown in Part I in discussing the thermal efficiency of the steam ship—that a huge proportion finds its way into the circulating water, and that only a small amount of this heat, considered as a proportion of the total fuel heat, actually appears as additional net work. In the Still experimental engine, this latter percentage was 4·2 per cent., although it formed actually some 15 per cent. of the effective work. Solid injection certainly does show a very distinct saving of power compared with air injection. The air blast power in Camellaird-Fullagar experimental engine represented 3·54 per cent. of the total heat expended, and one might reasonably assume that, had solid injection been adopted as in the Doxford engine (see Table IV), the brake efficiency would have been increased from 32·7 to, say, 35 per cent. at least; that is to say, by some extra 7 per cent. For the Armstrong-Sulzer engine alone is the anticipated exhaust gas temperature given 160° F. The Camellaird-Fullagar experimental showed about 430 deg. Such temperatures as these offer no scope at all for regeneration; they are very low for boiler funnel temperatures, even with air heating in the uptakes. It is a debatable point as to whether in the four-cycle engine, with its higher exhaust temperature, regeneration is advisable, though in their proposal, Vickers apparently consider that it is. This reduction in exhaust gas temperature shows a marked advance on the practice of a few years ago. It should be remembered, in considering the thermal losses, that a reduction of the cooling water loss may not necessarily mean a reduction in total thermal loss as the

temperature of the exhaust will rise correspondingly. So much valuable information has been collected in these proposals that it seems ungracious to suggest that just a little more would have been most welcome—exhaust temperatures or clearance volumes, for instance. The analysed comparative data certainly provides ample food for reflection

Comparative Losses.

In Fig. 3 is given a diagram in which the average distribution of heat losses in a steam ship are compared with those in a motor ship. In view of the varying proportions of thermal and mechanical losses in the latter due to difference in type—mainly, really, due to method of injection—the motor ship

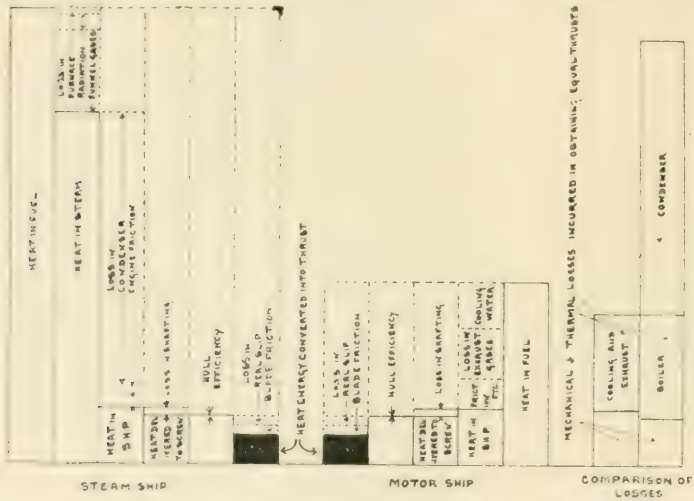


Fig. 3.—Comparison of Average Distribution of Heat Losses in Steam Ship and Motor Ship.

losses must be taken as approximate. Attention, however, must be directed to comparison of losses in the two cases. The frictional and mechanical losses to produce the same thrust are but slightly greater in the motor. The heat losses in the two cases are almost identical as between the loss in the boiler section and the cooling and exhaust of the motor. The really big loss in the steam ship lies in the condensing plant which is, practically, entirely over and above the whole sum of the losses in the motor ship, both mechanical and thermal.

The Prospects of Improvement.

The question naturally arises as to whether there are any immediate prospects of improvement in either method of propulsion. In the case of the steam ship it can only lie in the engine or the turbine. Boiler efficiencies of 85 per cent. are very nearly realisable in service with oil fuel and air heating; experiments on propellers and propeller position do not appear to give much grounds for hope of improvement in the external section, and, in any case, the loss in this region is comparatively [illegible] propelled by machinery of 6,000 h.p. and under that the internal-combustion engine seems destined to replace steam to a very much greater extent than that by which the turbine ever [illegible] power of turbine machinery afloat, but the overwhelming majority of it is installed in ships for which reciprocating plants [illegible] engines ever built were those of the present *Mount Vernon* [illegible]

is less than is fitted in many large torpedo craft to-day. The propeller arrangements of motor ships do still, we believe, leave something to be desired; with the wish to restrict engine sizes the revolutions per knot have been increased considerably above those of steam practice with the result that finer pitch ratios than would otherwise be adopted have been used. This, however, is only a passing phase, and the tabulated data shows clearly that some builders are prepared to come down to quite low revolutions. The proposed system of a Diesel-electric drive certainly removes some of the difficulties in the way of the design of larger units, but it introduces others, and, what is of still greater importance, it reduces the economy by the introduction of both dynamo and motor losses—a matter of some 8 or 10 per cent. Solid injection possesses several advantages besides the reduction in mechanical loss. Lower compression pressures can be used than with air blast on account of the cooling effect of the expanding injection air, and hence distinct reduction in certain engine scantlings can be effected. The theoretical efficiency of the cycle on which the Diesel engine works is given by the expression—

$$\eta = \frac{(\theta_4 - \theta_1)}{\gamma(\theta_3 - \theta_2)}$$

where the temperatures θ have their respective values at the points shown in the attached sketch. In the practical engine the conversion of the energy of the explosion into work on the piston is performed with a very high degree of efficiency indeed; so high, in fact, that there is little prospect of improvement here. We are, therefore, largely left with the problem of attaining the desired temperatures with the minimum amount of mechanical effort and thermal loss, and in the search for these, the nature of the cycle and the system of injection will be strenuously investigated for a long time to come.

It is precisely this fundamental reason of the high efficiency of the conversion of the energy in the gas which renders the probability of competition arising from any development of the internal-combustion turbine so very doubtful for many

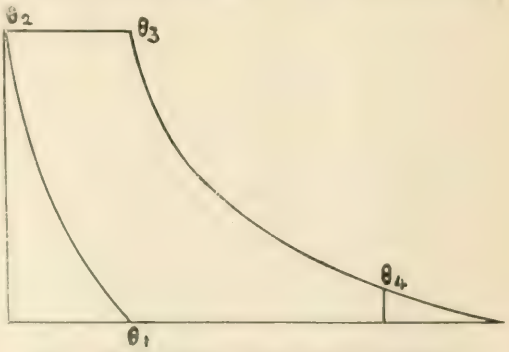


Fig. 4.—Sketch showing Temperatures mentioned in Efficiency Expression above.

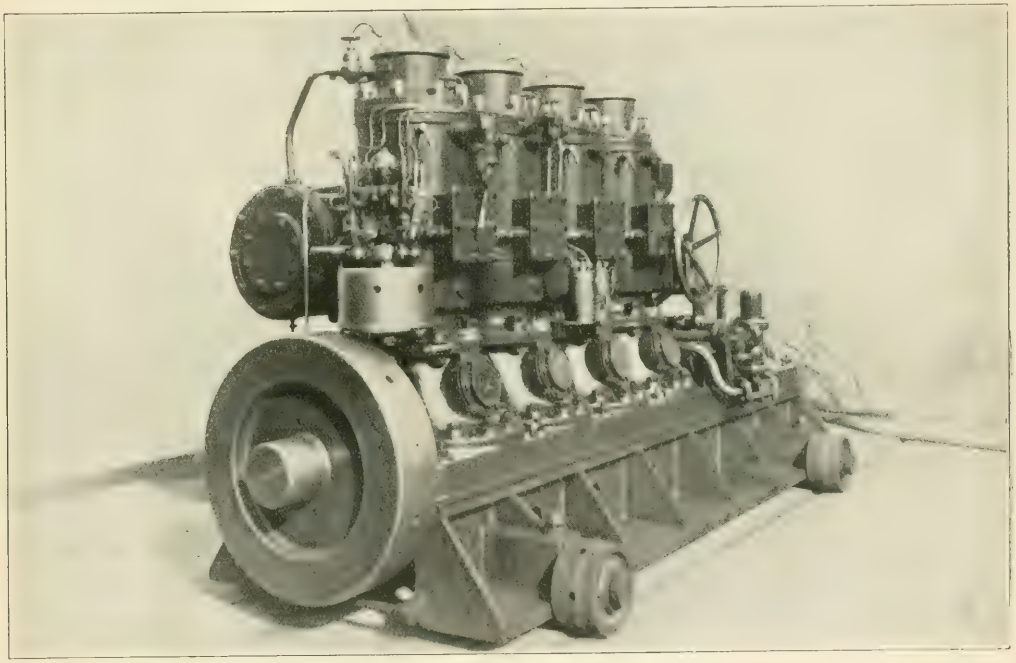
years to come, even if the temperature difficulties can be negotiated. It is not possible in the gas turbine to work with the vastly increased range of expansion which is possible with steam, owing to the change of state which the latter undergoes, and consequently the gas turbine cannot, as does the steam turbine, make up for its lower efficiency of energy conversion

(*Continued on page* 227.)

THE "BRITISH KROMHOUT" MARINE OIL ENGINE.

180/200 b.h.p. Unit, manufactured at Southampton.

We illustrate below a British Kromhout marine oil engine of 180/200 b.h.p. This internal-combustion heavy oil engine, which has been manufactured in England for the last 20 years, under a licence held from the patentees, by Perman & Co. Limited, 82/83, Fenchurch Street, London, E.C. 3, is now being manufactured by Day, Summers & Co. Limited, of Northam Iron Works, Southampton, under an agreement with Perman & Co. which was made about three years ago.



The 4-cylinder 180/200 b.h.p. 4 M 4 type British Kromhout Marine Oil Engine.

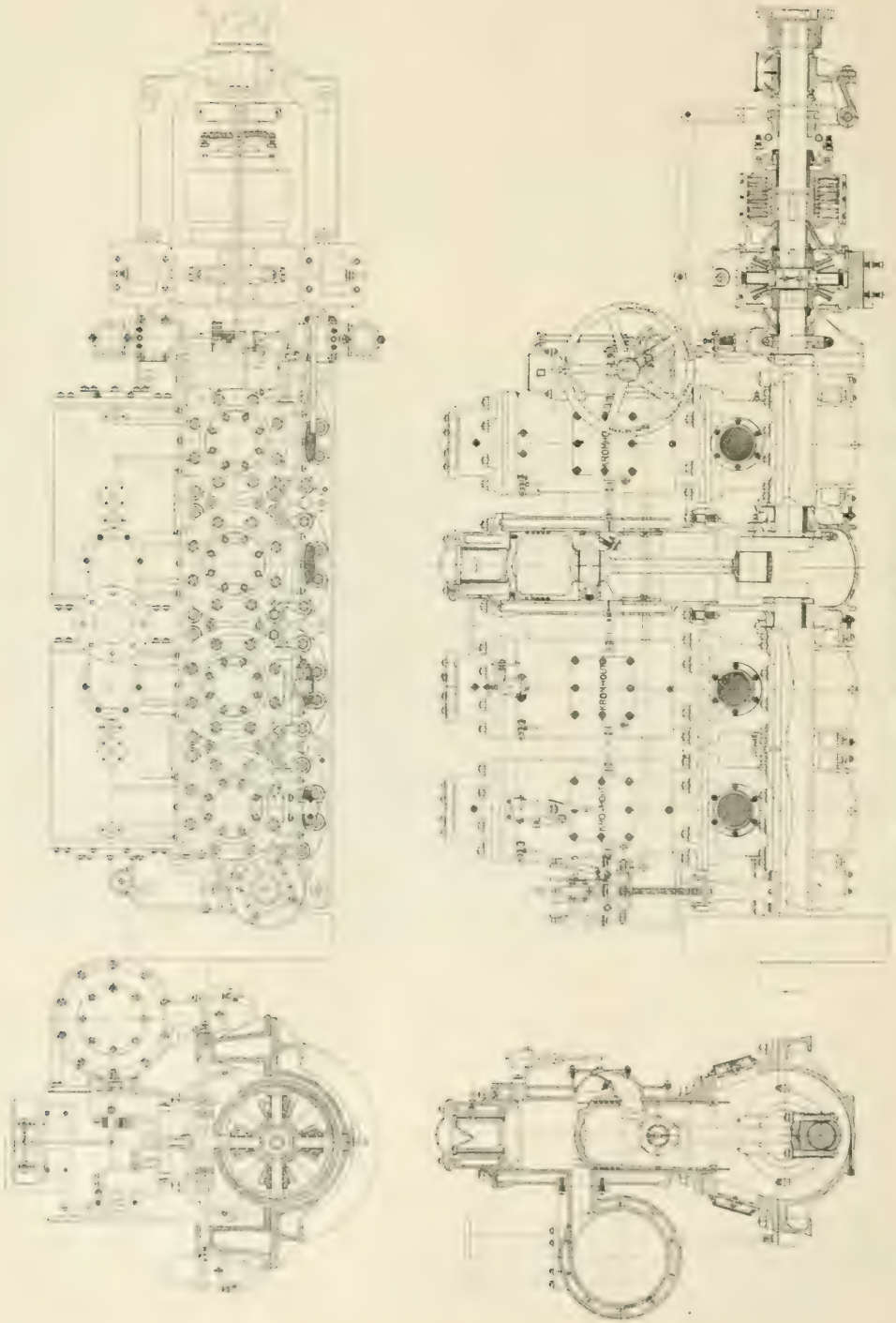
The engine illustrated in the photograph is of the 4-cylinder type, developing 200 b.h.p. when running at 320 r.p.m.

This engine is one of three sets supplied to the Anglo-Persian Oil Company for installation in oil tankers built by J. Samuel White & Co. Limited, of Cowes, and it is controlled by the handwheels shown at the after end, there being no other levers or control-wheels. This motor is non-reversible, but is fitted with the special "Kromhout" reverse gear which has given satisfaction for many years past.

Starting is effected by means of compressed air, at a pressure of about 200 lb. per sq. in., carried in reservoirs which are charged from the engine itself when running. The engine is started and runs slowly with the clutch "out." When it is necessary to get the vessel under weigh, by a turn of the control-wheel the engine is speeded up, and by the same motion the clutch, which is connected with the propeller shaft, is engaged. To increase or decrease the speed it is only necessary further to turn this same control-wheel to left or right.

This is effected, without disengaging the clutch, by having a parallel portion on the clutch so that the disengagement only takes place when the wheel has been turned sufficiently far for that purpose.

The control of the speed by the handwheel shown is effected by a cam on the handwheel which controls a horizontal shaft running the whole length of the engine. This shaft by lever and rod controls the tension on the springs of the governor, which is fitted in the drum shown in the foreground of the photograph immediately over the flywheel. The governor is worked by a vertical shaft driven off the crankshaft, and carries a continuous track cam on which run four rollers which operate, in turn, the fuel pumps supplying fuel to each cylinder. By the tapered form of the cam, which rises and falls in accordance with the speed of the engine, the stroke for the fuel pump varies, thus automatically reducing or increasing the supply of fuel to the cylinders. In addition to this, the same wheel controls the scavenging air supply by "butterfly" valves fitted behind the square doors shown on the front of each cylinder. When the engine is running at full speed, these valves are wide open. As the speed is reduced by turning the hand-wheel, the "butterfly" valves are closed and the supply of



General Arrangement Drawings of a 4 M 4 Krombout Marine Oil Engine.

scavenging air to the cylinders is automatically reduced. Reciprocating type bilge and circulating-water pumps are fitted to the engine—one on each side—driven from an eccentric on the crankshaft. One of these pumps may be seen in the photograph to the right of the handwheel.

The "Kromhout" engine is not a "hot bulb" engine, but is fitted with a special form of ignition plate at the cylinder top. To heat up this ignition plate before starting the engine, a special rapid heater nozzle is fitted as shown in the photograph. This nozzle is carried on a copper air-pipe, inside which is a smaller tube supplying paraffin. The intense heat from this blow lamp rapidly heats the ignition plate and the engine can be started up in about five minutes from stone cold. The compressed air for starting the engine is connected with the forward cylinder only, the piston of which has to be brought to the correct position first. The engine is then started by pulling the lever shown on the forward cylinder.

The two standard patterns of the "Kromhout" engine in which Day, Summers & Co. have specialised, are the M 3 and M 4 types. The M 3 type develops 35/40 b.h.p. per cylinder, while the latter type develops 45/50 b.h.p. per cylinder. The engine illustrated in the photograph is, as already stated, the 4 M 4 type. The firm are now developing the 6 M 6 type which will give a b.h.p. of 500. Two sets of 100 b.h.p. "Kromhout" engines, built some time ago, which were fitted in a yacht, have just been opened up after a 1,800-mile cruise and found to be in perfect condition. The firm have also built a considerable number of single-cylinder and two-cylinder engines of both the M 3 and M 4 type during the past two years, all of which are reported to have given the utmost satisfaction.

REFRIGERATOR BARGES.

A valuable report has been published recently by the Engineering Committee of the Food Investigation Board. It deals with experiments which have been carried out in the Thames district with a view to ascertaining the most suitable appliances for lighters, barges, and similar river craft engaged in the transport of frozen produce from ship to quay or cold store. In this country, as is well known, insulated non-self-propelling barges are used to transport produce from ship to quay or cold store, but on these vessels no attempt is made to pre-cool the cargo spaces, and the only refrigeration possible in addition to their insulation is the cooling effect of the refrigerated cargo itself, the temperature of which consequently rises in transit. The Committee considered that, in view of the advantages offered by water transport for the carriage of frozen produce not only on the Thames but also in other places, the problem should be thoroughly investigated and that it should be extended to self-propelled vessels fitted with refrigerating appliances; the fitting of apparatus in these enable pre-cooling to be applied independently of ice or of outside refrigerating establishments, and also render it possible also to apply cooling to the cargo itself if the transit occupies more than the comparatively short time usual on the river.

The tests were carried out on a Ministry of Shipping barge which was built originally for use on French and Belgian canals, and which was self-propelled. The vessel was longer than the ordinary river barge, but her length was advantageous, in that it permitted of the subdivision of the hold into three practically equal compartments in each of which different kinds of thicknesses of insulation could be tested simultaneously. The barge had a paraffin engine installed right aft for propelling purposes, and the propeller could be disconnected and the engine used to drive an electric generator. A refrigerating machine working on the ethyl chloride principle was installed forward, motor-driven, the motor obtaining its current from the electric generator. Actual refrigeration was carried out by means of brine circulation, each of the holds having two separate circuits, one circuit for overhead distribution, and the other for the sides and ends. The tests were carried out in hot and in cold weather, and they covered a variety of conditions of pre-cooling and refrigeration, and the results are interesting and important. They prove beyond doubt the value of pre-cooling, and show that the insulation in the experimental barge was efficient for the services in which she was engaged, *i.e.*, short runs from the docks to riverside stores; the tests showed, moreover, the possibilities of the water-borne transport of frozen produce on inland and coastal waterways, and incidentally the utility of the internal-combustion engine for such vessels as the barge under consideration. Barge and canal traffic has never been as important in this country as in France, Belgium, Germany and Holland, in spite of the efforts which have been made from time to time to revive it, but the type of vessel represented by this refrigerator barge has distinct possibilities on any canal system, and it will be interesting to see whether the matter is taken up seriously in a commercial way. The handling of perishable foodstuffs is a matter of great importance, and one, moreover, to which too little attention is directed, and the refrigerator barge seems at last to be a step in the right direction.

THERMAL AND MECHANICAL EFFICIENCIES IN STEAM AND MOTOR SHIPS—II.—(*Concluded from page* 224)

by working on a much more extended cycle, *i.e.*, by using the toe of the diagram, because there is very little "toe" to use. There are many cases, of course, in which a gas turbine driving dynamo and motors might be applied because of the enormous reduction in weight and space that they seem likely to offer, but as the expansion ratio can differ but little from that employed in the engine, the thermo-dynamic efficiency will be very much less indeed. At present the thermal efficiency ratio of the best steam turbine is some 75 to 77 per cent., and reciprocating oil engines have frequently exceeded 80 per cent. The most optimistic supporter of the gas turbine need not expect to obtain more than 60 to 65 per cent. when working on a constant volume cycle and very much less when working on a constant pressure cycle, so that from the point of view of efficiency although the gas turbine will be much superior to the steam engine, it does not appear in the least likely to effect the marine oil engine in the way in which the steam turbine displaced the reciprocator in certain classes of ships. No very rapid improvement in the brake efficiency of the oil engine seems at all likely in practice or theory at an early date. It must be remembered, as we have shown, that already an enormous stride has been accomplished *in more than doubling the thermo-propulsive efficiency of ships* in recent years by means of the introduction of the oil engine, and the possibility of future developments in this direction is clearly illustrated in the diagrams published in Part I of this article.

THE NEW BRITISH INDIA MOTOR LINER "DALGOMA."

Trials were carried out at the end of last month of the new British India liner *Dalgoma*, built and engined by Alex. Stephen & Sons, Ltd., Linthouse, Glasgow.

The *Dalgoma* is a twin-screw motor vessel of about 6,000 tons, propelled by Stephen-Sulzer four-cylinder engines, developing a collective shaft horse power of 3,200 at 85 r.p.m. All auxiliary machinery, including the deck machinery, is electrically driven, and for this purpose two Stephen-Sulzer auxiliary Diesel engines, each developing 110 b.h.p. at 200 r.p.m., are installed to drive the electric generators. The scavenging air for the main engines is supplied by an electrically-driven turbo-blower situated in a recess at the aft end of the engine room. Two blowers are provided, one of which is sufficient to supply the necessary air. The air can be drawn either from the deck or from the engine room as required. This is the first ship built in Great Britain in which this system of scavenging has been used. It results in a very easy regulation of the scavenging air pressure, a considerable shortening of the main engine, and a lower consumption of power to produce the necessary scavenging air.

All manœuvring gear for the main engines, together with the main switchboard and air control valves, are arranged on the top platform which thus becomes the starting platform, leaving the bottom platform quite clear and greatly improving the access to the crankshaft and other moving parts. Sea water is used for cooling the jackets and covers, and the cooling of the pistons can be either by salt water or fresh water as desired. A workshop is arranged on the 'tween decks at the port side, and a harbour lighting set, driven by a two-cylinder hot bulb engine, is situated on the starboard 'tween deck. The other machinery situated at the top platform level includes a vertical boiler for heating the oil in the tanks and for supplying steam to the evaporator, four starting air receivers, and a piston water cooler. The auxiliary machinery on the bottom platform includes two centrifugal jacket-cooling pumps, two centrifugal piston-cooling pumps, centrifugal bilge and sanitary pumps, a ballast pump, spare lubricating pump and piston-cooling water transfer pump, two auxiliary air compressors, two oil fuel transfer pumps, and an emergency bilge pump. There is also a centrifugal oil separator for dealing with the lubricating oil, and an electric heater for heating the oil before passing it through the separator.

On the bridge deck there is a small emergency lighting set and emergency compressor, and a small refrigerating plant and fire-extinguishing plant are situated in compartments further forward. Silencers for the main and auxiliary engines are situated at the base of the funnel, which is used as an upcast ventilator for the engine room.

During trials a speed of 13 knots was attained on the measured mile, and reversal from ahead to astern occupied about 12 seconds. Under dead slow conditions the engines ran steadily with two cylinders only on fuel at about 30 r.p.m. There was a total absence of smoke in the exhaust and a marked lack of vibration.

The whole of the main machinery and of the two 400 b.h.p. auxiliary engines were built by Alex. Stephen & Sons at their Linthouse Works, the castings being supplied by A. F. Craig & Co., Paisley.

A New Motor Lifeboat.

On Saturday, May 26, the second lifeboat to be fitted with the new type of engine designed by the Royal National Lifeboat Institution, and described in full on pages 19–28 of our January issue, was delivered safely at her station, Cromer, Norfolk. The vessel left the yard of her builders, J. Samuel White & Co. Ltd., Cowes, on May 23, and did the journey to her station in easy stages. On the first day she ran from Cowes to Newhaven, a distance of 54 miles; the second portion of her journey was from Newhaven to Ramsgate, a distance of 77 miles; the third from Ramsgate to Gorleston, 80 miles; and the last stage, consisting of the 30 miles or so from Gorleston to Cromer.

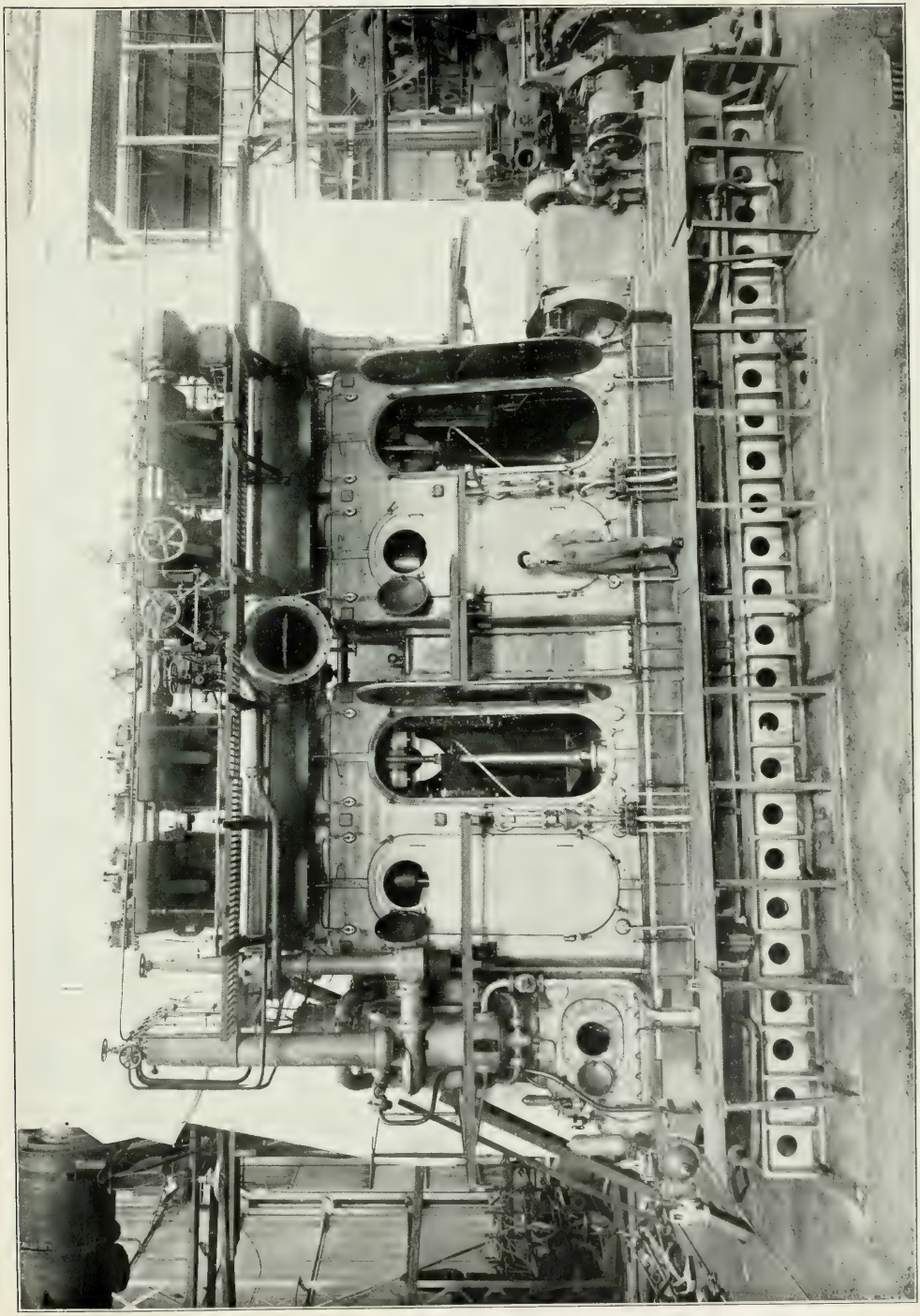
The attached table is based on figures obtained from the official log dealing with the first three days' run, and it is very interesting to compare the consumption of fuels for the various stages of the run. The number of gallons consumed per mile varies between four and five and a-half gallons, the number consumed per hour being between ·5 and ·6 of a gallon. With regard to the speed, it is interesting to note that an average of 9 knots was maintained on the first day, when running with a strong, favourable wind. The engine, throughout the complete run of four days, gave every satisfaction. The manœuvring qualities of the engine are particularly good, and in the first practice launch from the slipway the boat got away at practically full speed on touching the water.

EXTRACT FROM LOG OF NEW MOTOR LIFEBOAT.

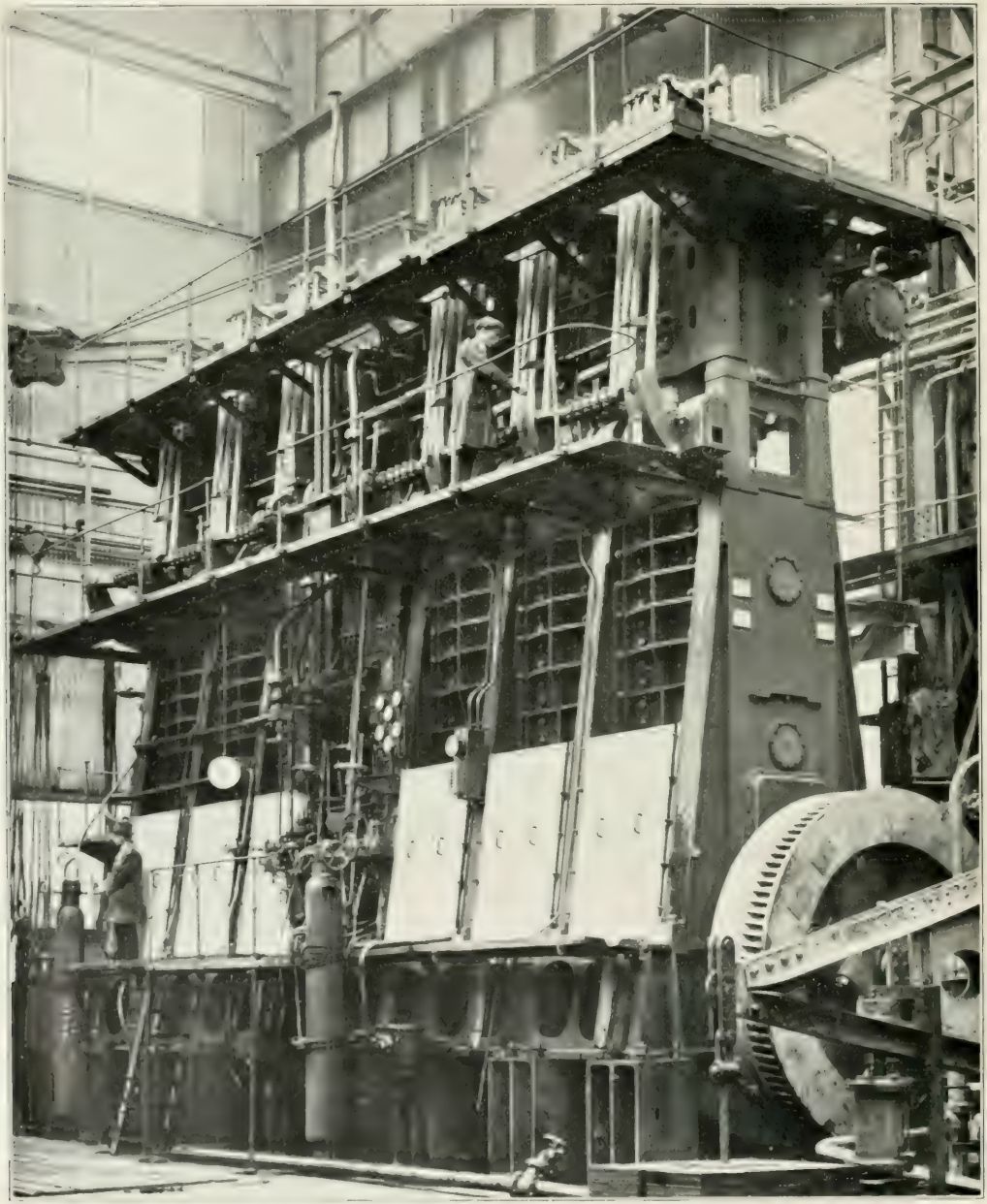
Particulars of Engine.—80 b.h.p., vertical, water-cooled, totally enclosed type; 6 cylinders; 5½-in. bore × 7-in. stroke.

Date.	Starting and finishing times.	Hours taken.	Distance in nautical miles.	R.P.M.	Average speed.	*Fuel consumed.	*Fuel consumption: (*a*) per hour, (*b*) per mile.	Remarks.
May 24, 1923	9.00 a.m. 3.00 p.m.	6	54	650	9	33 galls.	(*a*) 5·5 galls. (*b*) ·61 gall.	Set sail in Solent. Strong favourable wind.
May 24, 1923	7.00 a.m. 6.00 p.m.	11	77	700	7	46 galls.	(*a*) 4·18 galls. (*b*) ·59 gall.	12.55 p.m. off Dungeness, 3.55 p.m. off Dover. Cold; short choppy sea.
May 25, 1923	6.15 a.m. 4.00 p.m.	9·75 hours.	80	650	8	40 galls.	(*a*) 4·1 galls. (*b*) ·50 gall.	3.00 p.m. Lowestoft. Strong wind.

Average fuel consumption per day = *39·75 gallons. * Worked-out figures.



One of the Main Engines of the new British-India Motor Liner "Dalgoma," built by Alex. Stephen & Sons, Linthouse. (See page 228.)



The Burmeister and Wain-Harland & Wolff 1,800-b.h.p. Long-stroke Diesel Engine, built by John G. Kincaid & Co. Ltd., Greenock, for the "Arantza Mendi." (See page 220.)

THE CLOSED FEED SYSTEM.

Removal of the Condensate from Condenser and Discharge to Boilers without exposure to Atmosphere.

A closed feed system enables the condensate to be removed from the condenser and discharged to the boilers by rotary pumps without exposure to the atmosphere. Although there is a certain amount of doubt in the minds of marine engineers as to the desirability of installing small turbine-driven rotary pumps on steamships, the makers of the closed feed apparatus described below offer to shipbuilders desiring to adopt this method of boiler feeding a compact and satisfactory arrangement. One of the most important features in a closed feed system is the maintenance of a uniform pressure on the suction side of the rotary feed pump. For this purpose it is necessary to provide apparatus which will always ensure a quantity of water in the closed feed line sufficient for the requirements of the boiler feed pump. The Contraflo closed feed system has been designed with the view to maintaining the highest possible vacuum in the main condenser, and returning the condensate to the boilers without exposure to the atmosphere. The auxiliary exhaust steam is utilised to the utmost extent for feed heating purposes, and the resulting water of condensation is delivered into the vacuum system so that the air is removed from the water. The condensate is removed from the main condenser by a turbine-driven extraction pump which delivers through a non-return valve into the pipe A, and thence through the condensing tubes of the air ejector unit into the suction branch of the turbine-driven feed pump. This feed pump discharges the water through the main surface heater to the boilers. The vacuum in the main condenser is produced by means of a specially designed air ejector having two steam jets operating in series, the steam from the jets being condensed on the tubes through which the extraction pump discharges, the air passing to the atmosphere through the pipe D. The air ejector thus forms a small auxiliary heater, the heat in the steam used in the jets being absorbed by the feed water. Exhaust steam from the engine room auxiliaries passes into the main feed heater, where it gives up its heat to the feed water, the drainage being led by the way of pipe E into the air ejector, in which the greater portion of the heat in the drainage water is absorbed by the feed water. The drainage from the air ejector passes through the filter into the feed tank. In Fig. 1, water from the feed tank, and make-up water from the reserve tank, is drawn through the pipe F into the main condenser, so that such water is deærated in the condenser and is afterwards removed, together with the main condensate, by the extraction pump, which delivers into the feed system in the manner previously described. Any excess of water in the feed system passes through the relief valve into the feed tank. There is provided an automatic by-pass valve, which opens and by-passes water from the feed tank into the condenser through the pipe F when the load on the main turbine falls below a predetermined amount. Thus, when little or no steam is condensed in the condenser, for example, when leaving harbour, water is automatically circulated through the condenser in sufficient quantity to maintain the desired pressure in the feed line, and for condensing

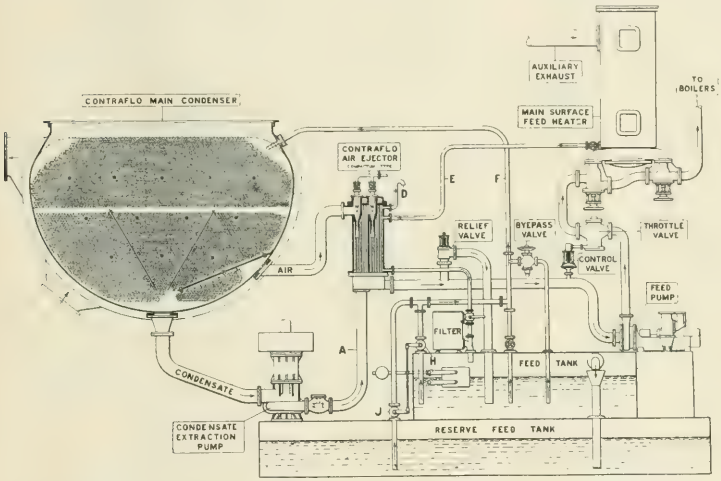


Fig. 1.—The Contraflo Closed Feed System. Diagrammatic Sketch of Arrangement.

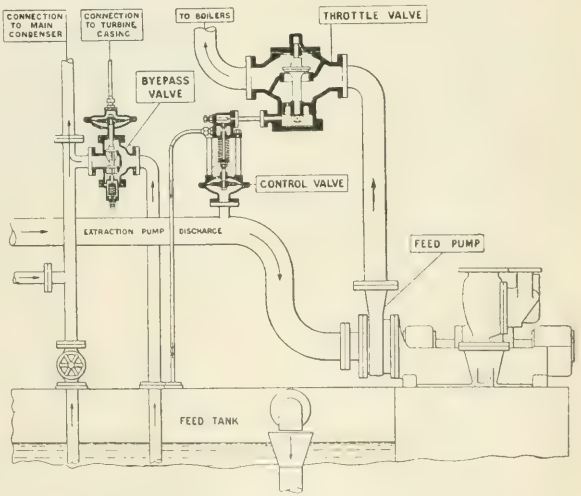


Fig. 2.—Arrangement of By-pass, Throttle and Control Valves.

steam to the air ejector. This by-pass valve, together with a throttle valve and a control valve, are illustrated in Fig. 2. Water from the feed tank is admitted into the feed system through the float controlled valve H which is opened by the float when the water level in the feed tank rises. When the

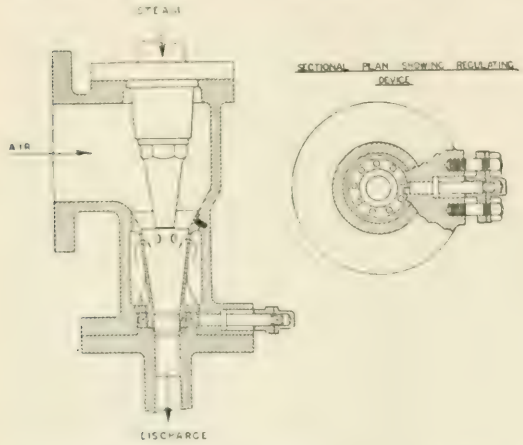


Fig. 3.—The Contraflo Closed Feed System. Stabilising Device.

water in the feed tank falls below a predetermined level the valve H is closed, and the float opens the valve J, so that water is drawn into the vacuum system from the reserve feed tank. Thus at all times a sufficient supply of water is provided, the amount passed through the valve J being more than that required by the feed pump. In that there is a surplus which passes through the relief valve and fills the feed tank to the normal level.

In a closed system having rotary pumps it is necessary to provide means for maintaining a constant circulation through the system when the condensate from the condenser ceases or is reduced, as previously mentioned, or when the turbines are running intermittently, and Fig. 2 shows in detail the arrangement of by-pass valve and control valve adopted in Fig. 1.

By-Pass Valve.—To provide for a constant circulation in the system, water is by-passed from the feed tank into the main condenser whenever the power on the main turbine falls below a predetermined load. This is accomplished by an automatic by-pass valve controlled by a diaphragm operated by the pressure in the casing of the low-pressure turbine. When the turbine is stopped or under light load, the by-pass valve is held open by its spring, and water passes through the valve into the condenser; but when the load on the turbine increases, the steam pressure acting on the diaphragm overcomes the spring, and thus closes the by-pass valve, so that the return of water to the condenser through this valve is shut off, the valve remaining closed until the turbine is again under light load. By this means a constant circulation of water through the system is provided, so that under all circumstances there is a sufficient supply of water for condensing steam and cooling the hot drainage in the air ejector.

Throttle and Control Valves.—It is recommended that the throttle and control valves shown in Fig. 2 should be used to prevent any possible trouble with the rotary feed pump in the unlikely event of a shortage of water occurring in the extraction pump discharge line. The throttle valve consists of an ordinary non-return valve attached to a piston which has a small hole for the escape of water. This leakage of water is regulated by the control valve which is operated by the pressure in the extraction pump discharge line. In the event of the pressure in the extraction pump discharge line falling owing to insufficiency of water, the control valve is immediately opened by the slight fall in pressure; the water below the throttle valve piston is released and the throttle valve closes until the water supply on the suction side of the feed pump is restored.

The Contraflo Stabilising Device.—Fig. 3 shows the device. When the quantity of ærated vapour discharged through the throat of an air ejector falls below the quantity for which the ejector is designed, the apparatus becomes unstable. In this system of stabilising, a quantity of vapour is returned from the throat of the ejector into the entrance cone. This by-pass takes place automatically whenever additional vapour is required, and instantaneously increases the stream in the throat, thus ensuring the stability of the ejector under all conditions of working.

Several ships have been fitted with elevated feed control tanks, because some superintendent engineers prefer a definite static head of water on the suction side of the feed pump, and an arrangement of this kind is shown in Fig. 4. The arrangement of the plant is very much the same as that previously described, except that the elevated tank in Fig. 4 takes the place of the relief valve shown in Fig. 1. The level of the water in the elevated tank, and consequently the head or pressure in

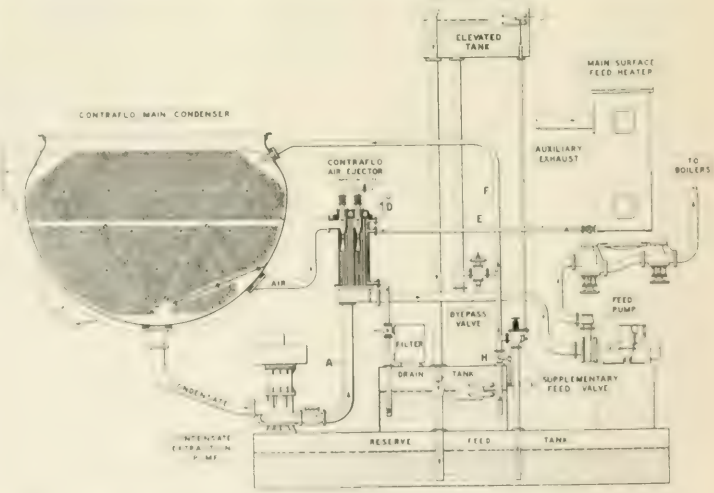


Fig. 4.—The Contraflo Closed Feed System with Elevated Feed Control Tank.

the extraction pump discharge line, is maintained by means of the float in the elevated tank which controls hydraulically the supplementary feed valve which admits water from the reserve feed tank into the main condenser through the pipe F. Any excess of water in the elevated tank overflows into the drain tank.

MARINE DIESEL ENGINES—RUNNING CONDITIONS AND OPERATING SYMPTOMS*—IV.

Starting Momentum—Position of Control Platform—Auxiliary Engines.

(*Concluded from page* 114.)

Another point, although bearing indirectly on the question of conservation of air, is, nevertheless, an important one; that is, starting momentum.

When an engine is being started in answer to a telegraph order for slow speed, it is quite usual to see the revolution speed rising momentarily to an abnormal degree, and then, when all fuel valves are in operation and the engine has settled down, the speed dropping to a steady figure, say, 40 r.p.m. The reason for this irregular acceleration is due to some fuel having been trapped in the injection valves as the manœuvring gear was run into neutral when previously stopping the engine. This fuel, plus an additional quantity supplied by the pumps when starting, is injected into the cylinders as the manœuvring gear passes from one position to the next with the result that the revolutions rise abnormally until a position is reached, when the speed drops due to the effect of reduced fuel valve lift. The effect of this is to give a powerful impulse to the ship, with the result that the order is curtailed to a very short duration, should the ship be docking at the time. Since it is during docking operations that the most frequent demands for engine movements are received, and, therefore the greatest need for the economical use of air exists, it is highly desirable that the starting momentum should be reduced to a minimum so as to prolong each movement to maximum duration, thus providing an opportunity for the engine to make up some of the air that has been used in starting.

Control Platform Position.

At this juncture it is interesting to consider the question of the most advantageous position for the control platform on a Diesel engine.

In the majority of cases the control gear is arranged on the bottom platform level, following standard steam practice and tradition, but the arrangement is open to criticism where top-level camshaft engines are concerned. When this type of engine is specially considered, it will be found that most of the gear deserving the closest attention is arranged at the top of the engine, while at the control platform attention is confined mainly to gauges. Were the controls arranged at the top-platform level, then all the valve gear would be under more constant supervision, since it is on the control platform that most of the time on watch is, of necessity, spent. Again, the fuel settling tanks are almost invariably arranged on the top platform, and these would be more readily checked and handled when filling. Another advantage lies in the fact that it is possible to obtain a more comprehensive grasp of the action of the engine when manœuvring, which can be made to result in a saving of starting air—a very important point.

Top Platform.

If it is argued that the top platform is not neglected with the present arrangement, then it can be equally assumed, that were the controls arranged on top, the rest of the engine would not be neglected.

While auxiliary engines are usually arranged on the bottom platform, this would not matter, since, even with the present arrangement, they are not under constant supervision, while the same indication of their running, apart from actual examination, exists equally well at the top-platform level.

It will be found that in many of the modern double-reduction steam turbine engine rooms, the control platform is raised to the turbine level, while the air, feed, and circulating pumps are arranged under it and out of sight. Even the dynamo sets, in cases where most of the auxiliaries are electrically driven, are also far removed from the control platform, yet the arrangement would appear to be entirely satisfactory.

What applies to the steam-engine room as quoted, is equally applicable to the engine-room of a motorship, while the question resolves itself into placing the control platform, if at all possible, at the place where most supervision is required. For a Diesel engine with top-level camshaft, the top platform is undoubtedly that place.

Auxiliary Engines.

Much of what has been stated regarding main engines is applicable to auxiliary engines, but there are other factors preventing the application as a whole.

Since there is at least always one auxiliary engine as stand-by, maintenance work is rendered easier, while the engines can always be kept in a state to allow of full output being developed when required.

Air testing of the cylinders and valves is even more necessary in the case of auxiliary engines if the exact condition of each part is to be known, as indicator cards taken from these engines are even less reliable than main-engine cards, owing to the higher speed and greater inertia effects of the indicator.

Again, auxiliary engines are operated under rather more adverse conditions than main engines, this being caused by varying loads and the difficulty of maintaining perfect combustion under each. In view of this, it is advisable only to run the engines for a comparatively short time—say, four days at a stretch—until the service time for each constituent part is established.

Working Cargo.

The worst conditions in respect of combustion are met with when loading or discharging cargo, and, at the best, a compromise has to be effected to ensure reasonable operation throughout the range.

When the load is light, the blast pressure should be 600 lb. per sq. in. or under (dependent upon the compression pressures), while the fuel injection valve lift should be as small as possible, these conditions tending to maintain steady firing with combustion at constant pressure.

Heavy loads, on the contrary, require a high blast pressure and the fuel injection valve lift increased if the best results are to be attained, while anything in the nature of an overload necessitates a blast pressure of at least 1,000 lb. per sq in., in conjunction with the maximum fuel injection valve lift. Since cargo work involves loads ranging from zero to 120 per cent., neither extreme in adjustment is suitable, so that settings have to be arranged which will allow the engine to operate under reasonable conditions over the entire range. This is usually effected by keeping the blast pressure about 750 lb. per sq. in., and by adjusting the fuel injection valve lift to 75 per cent. maximum. With these settings the engine runs best at moderate load, but will take full load for short periods, while on light load continuity of firing is good provided the period is not too long. Of course, a variation of these settings is advisable

* Abstract of paper read by Mr. A. J. Brown, before the Institute of Marine Engineers.

working in the average load expected. On some occasions the average load might be on the light side, when reduced blast pressure and fuel lift valve would be desirable, while, on the other hand, a heavier average load would best be met with an increase of blast and fuel valve lift. The question becomes more important when two or more engines are run in parallel, some irregular running results on even loading, and may end in one engine cutting out when the load goes off momentarily, so that the remaining engine has to sustain the increase when the load again comes on.

The effect of high blast pressure, in conjunction with excessive fuel injection valve lift, during light loads, is to cause missfiring. The reason for this is due to the expansion of air through the fuel valve causing a reduction of temperature in the cylinder with the result that combustion is occasionally prevented, especially when the governor has reduced the quantity of fuel being delivered. The fuel, however, is partially vaporised and shows in the exhaust as white smoke, and the cylinder is never wholly scavenged. The result of this is, that over a stroke or two a combustible mixture is formed which finally explodes during a compression stroke, causing detonation and providing ignition for the following charge of fuel. This may again heat up the cylinder so that firing is regular for a few strokes until the governor again reduces the quantity when missfiring once more becomes evident.

Low blast pressure and reduced fuel injection valve lift during heavy loads shows in the exhaust as black smoke, this being caused through imperfect atomisation of the fuel, with the result that combustion is incomplete and the power is reduced. After-burning also takes place, which, in turn, reduces the weight of air taken in on the suction stroke, thus tending to lower the output still further.

In connection with the lubrication of auxiliary engines, attention should be paid to the condition of the oil in the system as it is more liable to contamination than is the lubricating oil in the main engines. The reason for this is that auxiliary engines being fitted with trunk pistons, all the burnt and carbonised oil passes direct into the crankcase. Again, unless the pistons are provided with extra long skirts, it will be found that a quantity of oil is splashed on to the underside of the piston crowns, and this oil also forms carbon which helps to contaminate the lubricating oil supply.

Cylinder lubrication is materially assisted by the amount of splash in the crankcase, to such an extent, indeed, that the extra supply given by the mechanical lubricators is almost unnecessary. This is amply proved by air compressors fitted with trunk pistons open to crankcase. Where these are concerned, the great difficulty is to prevent an excessive supply of lubricating oil passing into the compressor cylinder, and, if this is attributed to suction effect, then suction effect must also become manifest in the power cylinders.

Auxiliary Compressors.

Considering now the air compressors fitted to auxiliary engines; these really have to play a dual part, in that not only has each to supply blast air to its own engine, but, when the main engines are being manœuvred, it is sometimes necessary to draw upon the surplus air from them. This necessitates these compressors always being capable of full output when entering or leaving port, and this can only be assured by cleaning the valves, ports and passages at regular intervals. What has been stated in connection with main-engine compressors applies equally to those of the auxiliary engines. Where with the main-engine compressors a certain amount of carbonisation of valves, &c., is unavoidable, due mainly to the time that must elapse before they can be cleaned, there is, therefore, all the more reason why the auxiliary engine compressors should be kept in the best condition.

Auxiliary engine compressor valves will tend to carbonise more readily than those of the main compressors, due mainly to more lubrication being given throughout, the unwanted supply coming from the crankcase splash and being unduly increased by higher revolution speed and reduced clearance space in the crankcase. Another reason is that higher compression temperatures are involved through the compressors being, usually, of the two-stage type.

Then stage balance also plays an important part. It is usual to find the compressors capable of giving at least 25 per cent. more output than is needed for blast during normal loads. The problem, then, is to operate the compressors at 75 per cent. output and still maintain a good temperature balance. The usual procedure would appear to be to regulate the quantity of air supplied by throttling the L.P. suction intake, or by blowing-off a certain quantity of air after it has been compressed in the L.P. stage. Neither system is good unless the H.P. stage is balanced to suit. To do this, it is necessary to work with a higher L.P. pressure, while the surplus air can be discharged through the H.P. drains. Although this system absorbs more power in effect, the cost is amply balanced by a reduced maintenance bill.

The best temperature balance is achieved by working each stage at the same compression ratio, having regard to the efficacy of the intercoolers, but although this is not always possible, adjustments can be made to bring the balance within reason. Compression temperature troubles incurred through wrong stage balancing can be practically abolished by the provision of suitable thermometers in the ship's outfit.

Starting Auxiliary Engines.

With regard to starting auxiliary engines, what has been said of the fallacy of running the main engines with all cylinders on air once the engine is in motion, applies equally to the auxiliaries. With them, however, it is usually necessary to give more than one air impulse before introducing the fuel, unless the starting air pressure is very high. A greater momentum is necessary to ensure the compression temperature being sufficiently high. This is due to the fact that the smaller the cylinder the greater proportionately does the amount of cooling surface become, so that the minimum revolution speed is higher than in the case of a large engine.

It is sometimes advantageous, when starting with low starting air pressure, to allow the engine to make the first revolution or two with the blast air shut off, but with the fuel pumps in action, and then, as the first fuel injection valve is cut in, to open the blast air valve. Doing this approximates to solid injection, which has the advantage of readier combustion.

In a case where the blast air pressure as well as the starting air pressure is low, and it is not possible to recharge the reservoirs, the best method to adopt is to give a small injection of paraffin through each indicator cock before starting. The cylinder relief valves may lift, but the result is effective, although this method is not to be recommended as habitual practice every time an auxiliary is being started.

The trend in design of auxiliary engines appears to be towards a reduction in revolution speed, and it is a point that will be appreciated by those marine engineers who have sailed with auxiliary engines running at 300 r.p.m. and over. Unless in cases where expense has been no object, the reciprocating weights of such engines are usually excessive, resulting in great inertia load with consequent wear of bearings, especially top-half bushes. Such engines usually require all main and connecting-rod bushes to be overhauled and adjusted every six months, especially in cases where the lubricating oil has not been properly filtered and cleaned during that period.

In conclusion, it would be well to draw attention to the necessity for organised routine. No hard and fast rules can be laid down for the overhauling time of any parts, as it is all a matter of circumstance. Extra work will be essential at the start when dealing with new engines until such time as definite information is established, but, as already pointed out, such work will be justified by a reduction of maintenance costs.

NOTES FROM THE FOREIGN PRESS.

Specially abstracted for "The Marine Engineer and Naval Architect."

The Transatlantic Passenger Service of the Future.

The future of the Atlantic passenger-carrying trade is always an interesting speculation, and in this article Dr. Foerster gives a very clear exposition of the probable trend of the development. From the point of view of efficiency of propulsion, all of the largest existing Atlantic high-speed liners can be improved, either by the introduction of the geared turbine or by the Diesel engine. But apart from this, if we accept the proposition that a higher speed than at present is essential, conditions of convenience of arrival and departure times at London and New York demand a sea speed of 28·35 knots as opposed to the present 22 to 23 knots. Taking the *Majestic* as a basis, the author shows that a vessel capable of working at the higher speed would be 303 metres long on the waterline, the displacement would be 72,800 tons, the maximum horse-power (geared turbines) 165,000, and number of passengers 3,300. The next highest convenient service would necessitate a speed of 32 to 33 knots, a vessel 360 metres long, 107,000 tons displacement, horse-power 380,000. This further proposition is hardly practicable, let alone commercial. This limit to ocean transport, however, directs attention to the practicability of aircraft, and particularly of Zeppelin transport. The author outlines in this connection a design of Zeppelin for mail and passenger purposes, capable of flying from Berlin to New York at an average speed of 65 knots. By a very careful analysis of available mails, the author shows that carrying 40 passengers and 12 tons of mails for 100 trips per year, the proposition, after all allowances are made, should be quite a practical one. In the author's opinion the Zeppelin, once its safety is definitely established, will prove a serious competitor to the high-speed liner. From this point the author takes up the question of the most economical, as apart from the fastest, liner. Assuming the present service speed of 22 to 23 knots, he shows that, by using the *Vaterland* as a basis, the economy of geared turbines, cruiser stern, bulges (*i.e.*, the Foerster Form-stabile type), and, finally, the Flettner rudder, enables an extra deck of first-class passengers to be carried. This design has a length on the waterline of 275 metres, displacement 58,700, machinery horse-power 55,000, and carries 4,200 passengers. In concluding, the author expresses doubt as to the realisation of the "four days" ship in view of air competition, and emphasises the importance of the more economic design on existing dimensions and conditions of service.—*Dr. Foerster, Werft, Reederei, Hafen*, March 7, 1923; p. 114; 6 pp.; 10 ff.

Propeller Strength Similarity.

It is well known that the blade thickness of a propeller has a considerable influence upon this latter's performance. So much so that a propeller based upon some particular model screw cannot have its blade thickness changed with impunity to satisfy conditions of strength in the large propeller, and yet possess an identical performance with the model screw. The practical problem thus resolves itself into finding the blade thickness which the model screw should possess in order to give a safe stress in the full-size propeller. This step, in the author's opinion, should not have to be taken by the practitioner, but should be undertaken by the experimenter when publishing the results of his model screw research. To show how the experimenter may accomplish this the author investigates the variations of stress in similar propellers. The moments perpendicular and parallel to the blade as it enters the boss consist of the resolved moments due to torque and thrust. Expressions are deduced from these on an assumed shape of blade; similarly, the section modulus of the blade is approximated to. From these quantities an expression for stress is deduced, and, finally, a dimensionless expression connecting permissible stress, revolutions, diameter and shaft horse-power is derived. This expression, could be calculated for the model screw, and from it the suitability of the particular type for a given absolute size could be immediately realised without necessitating a detail calculation.—*Dr. C. von den Steinen, Werft, Reederei, Hafen*, February 22, 1923; p. 89; 2 pp.; 1 ff.

Directional Stability and Manœuvrability of Ships.

By directional stability is usually understood the power with which a vessel opposes any angular change in its direction of motion. It is usually measured in practice by carrying a certain angle of helm and noting the corresponding angular change in the ship's direction of motion. No forces are usually measured, and thus the usual information is but of little use to constructors for the designing of future vessels. The ordinary turning trials in which the tactical diameter is determined cannot be carried out in the case of one important class of vessel, *i.e.*, river vessels. Limitations of turning space necessitate the development of another method of testing the manœuvrability of these vessels; and in this article the author considers in a very comprehensive manner the theory of the manœuvring of ships. The usual complexity of the subject is considerably ameliorated by making use of the analogy of the ordinary initial transverse stability of ships. Thus a vessel has a positive directional stability when the centre of lateral resistance is aft of the vessel's centre of gravity; the separation between the two centres the author proposes to call the metacentric length of directional stability. From the equations of motion investigated several interesting conclusions are drawn. For instance, it is shown that the tactical diameter is independent of the distribution of weights on the ship and only depends for a given speed upon the vessel's form. This fact shows the potentiality of model experiments for predetermining the manœuvring qualities of various ship forms associated with various rudders. A knowledge of the model's drift angle, transverse and longitudinal resistances, rudder angle and pressure, enables the tactical diameter to be calculated. Curves of drift angle for each ship speed are plotted on a base of rudder angle and show up the manœuvrability of the model very clearly. Another interesting analogy is that corresponding to the ordinary curve of statical stability. The directional stability for a given speed is plotted on a base of drift angle as is also the turning moment of the rudder about the vessel's centre of gravity. This latter is analogous to the heeling moment due to wind pressure, and thus where the two curves intersect gives the balancing drift angle. Beyond this angle the vessel becomes directionally unstable. These curves of statical directional stability form a useful means of investigating the comparative manœuvrability of ships.—*Dr. Marcell Klein, Schiffbau*, May 9, 1923; p 505; 7 pp.; 10 ff.

The Resistance and Propulsion of Ships.

In a previous article (*see* January notes) the author dealt with the laws of comparison of the frictional and wave-making resistances of ships. In this article an approximate solution of the problem of deriving a vessel's horse-power from that of another vessel of identical lines but of shorter length is attempted. When the length ratio and speed ratio are each equal to λ, and assuming that frictional resistance varies as the 1·83 power of the speed, then the ratio of the frictional horse-powers becomes $\lambda^{3\cdot 13}$. From a piece of not too convincing mathematics the author presumes that the corresponding ratio of form or wave-making powers becomes λ^3. These ratios are applied first of all to the respective effective horse-powers; but assuming that the propulsive coefficient remains unchanged, the ratios are also presumed to hold good for the indicated powers. Examples showing the application of the method are given, the reasonableness of the results obtained being judged by estimating the power of existing longer ships from the data of smaller vessels.

When in lengthening the ship the speed is kept constant the frictional resistance increases and the wave-making decreases. At some point in the lengthening the total resistance will be a minimum, but whether this minimum is of any consequence is problematical.—*Geheimrat Rudloff, Schiffbau*, April 4/11, 1923; p. 442; 3 pp.; 6 ff.

Ship Trials and Model Experiments.

This article forms an interesting example of the successful use of model experiments both for the improvement of ship form and the correct assessment of ship power. Details of the twin-screw motor ship *Borodino* were given some time ago in *Schiffbau* (December 20/27, 1922), in an article by Blomerius describing the Volga river vessels. Very complete trial data of this vessel were included, and these data have enabled Dr. Kempf, Director of the Hamburg Tank, to make an interesting comparison between the *Borodino's* model experiments (which were carried out at the Uebigau Tank in 1910) and her trial data. The *Borodino's* principal dimensions are as follows:—Length,

so [illegible], breadth, 9·3 m., depth, 3·2 m., and draught, 1·60 m. To investigate the conditions of the Volga, model experiments were carried out at depths of water corresponding to 2 m., 4 m., 6 m., and deep water. It was hoped to obtain a speed of 13 knots with a 1,200 b.h.p. engine, which was by no means [illegible], but the experiments clearly showed that even at 6 per cent. the steaming effect was so pronounced that only a speed of 10 knots could be obtained with this expenditure of power. It was thus necessary to alter the after form of the ship, and when this was done the excessive change of trim and wave formation which accompanied the first form was greatly reduced and consequently the resistance. The experiments were also carried out with propellers attached, and one very important point in the powering of motor vessels was clearly brought out. The propeller finally adopted was not the most efficient, but because the machinery could only develop its full power at 240 revolutions it was essential to adopt a less efficient propeller in order to get the full power from the engine. The model experiment finally showed that a speed of 24·10 km. per hour could be obtained on 1,200 b.h.p. at 240 revolutions. The trial trip showed that a speed of 24·04 km. per hour was obtained on the same power and revs. Exact information as to the trial conditions were not available, and if one is tempted to assume that these did not differ greatly from those of the model experiment then the comparison shows very good agreement.—*Dr. C. Kempf*, *Schiffbau*, April 25, 1923; p. 484; 2 pp.; 2 ff.

Trim.

Much has been written by naval architects and others concerning the advisability of placing more stability information at the disposal of ships' officers, chiefly, of course, from the point of view of securing the safety of ships. Longitudinal stability, *i.e.*, trim, does not involve questions of safety, but there are good economical reasons for giving a ship's officer sufficient trim information to enable him to trim his ship in a manner which will ensure the least consumption of fuel. Generally this is accomplished when the vessel trims by the stern, and thus any information placed at the officer's disposal can eliminate a lot of trial and error on his part in securing a satisfactory trim. In this article, written, suitably enough, by a ship's officer, it is shown how, in the absence of exact information, the assumption that the ship's length is equal to the longitudinal metacentric height enables most trim problems met with in practice to be expeditiously solved. On this basis the author has developed an interesting although not entirely new presentation of the trim data. Tables are prepared giving at each draught the change in trim brought about by placing a weight of 100 tons at the after and forward portion of each hold. From these tables more complete tables giving the corresponding changes of draught forward and aft are prepared. A similar table for the water ballast tanks is also prepared. The author thus shows how even in the absence of exact data useful information can be prepared. With the co-operation of the builder the officer could ensure the trim of his ship without much arithmetical work, and as good trim means a saving of fuel consumption it is thus in the owner's interest to see that this information is placed at the officer's disposal by the builder. The author gives the tables prepared for the H.A.L. steamer *Dania* as an illustration of the method.—*A. Last, Werft, Reederei, Hafen*, May 7, 1923; p. 219; 2½ pp.

High Speed Indicators.

There are evident signs in many countries that economy in ship running can be very largely effected by close attention to details which a few years ago were considered fanciful. We refer here particularly to the use of measuring instruments of all descriptions [illegible]. Practically every by which an engineer was expected to sense economy are being realised as capable of great improvement, but the day when every engine room constitutes a power station run with the minimum waste is still afar off. The oldest measuring instrument is, of course, the indicator. For slow speed steam work there are many satisfactory types, but for high speed steam, and especially internal-combustion work, although reliable types do exist yet very few are sufficiently simple to receive general adoption in practice. The chief difficulty in high speed work is, of course, the inertia of the indicator parts and the difficulty of securing continuous contact between the indicator registering point and the indicator card. To eliminate these defects Dr. J. Geiger has developed a "Precision Indicator" which gives very good results on service and is at the same time very simple. By careful attention to the critical period of vibration of the indicator itself the dimensions of the mechanism are so regulated as to ensure this critical speed only occurring at above the revolutions for which the instrument is intended. Several examples of cards taken with the normal indicator and the new precision indicator are shown, from which is easily seen how simply inaccuracies in the registering of power may occur through even apparently well-designed indicators.—*Werft, Reederei, Hafen*, May 7, 1923; p. 233; 1 p.; 9 ff.

Model Experiments on Rhine Barges.

During and since the war many attempts have been made to simplify the form of ships in order to reduce the labour involved in their construction. Generally it was found that simplified forms could be produced without in any way increasing the power necessary for the propulsion of the vessels. Naturally this experience would influence barge builders who have a *prima facie* case for standardisation to adopt a simplified form of construction, and this was done with a fair measure of success by the Gütehoffnungshütte of Walsum. However, the design of Rhine barges which is specially considered in this article does not admit of too rapid a generalisation of the advantages of simplified form. These barges necessarily have a long parallel body and have to be towed in shallow water at a speed of 15 km. per hour. The Rhine barge, as produced from the accumulated experience of over 30 years, has proved very economical in service, and thus the simplified form has to have considerable saving in first cost to justify its adoption. To investigate definitely the relative advantage, model experiments were carried out on a normal model and simplified model of a barge of length 78 m., breadth 9·43 m., draught 2·50 m., and displacement 1,630 m³. The centre of buoyancy of the normal form was 1 m. forward of 'midships and the simplified form 1·5 m. From Zeyss's experiments on merchant forms the chine line forward was arranged to run from half draught at the stem, whilst aft the V-shaped sections ran off at an angle of 40 deg. to the base line. However, in spite of utilising the best available lessons, experiments carried out at depths of water equal to "deep water," twice the draught, once the draught, and one-fifth of the draught showed the general and definite superiority of the normal form. With the single exception of the smallest depth of water when for speeds up to 11 km. per hour the simplified form had from 3 to 5 per cent less resistance, the normal form for all higher working speeds had from 12 to 18 per cent. less resistance. At the lowest depth it is interesting to note that both models touched the tank bottom at a speed corresponding to 14·5 km. per hour. This is indicative to a sinkage of 0·5 m. The superiority of the normal form is brought out by considering the resistance at a depth of water of 5 m. and a speed of 12 km. per hour. Here the respective resistances are 2,015 kg. and 2,345 kg. A tug capable of exerting a pull of 6,000 kg. can thus pull three normal barges, but only two simplified barges at this speed. If three of the latter have to be towed the tug will need to exert practically 220 indicated horse-power more than when towing the normal barges at the same speed.

Very complete tank information accompanies the article, which in itself forms a valuable addition to our knowledge of resistance in shallow water.—*Dr. G. Weddigen, Hut Schiffbau*, May 4, 1923; p. 105; 5 pp.; 4 ff.

The Kempf Stability Indicator.

Much of the talk about giving ships' officers accurate stability information of their ships unfortunately but rarely passes the paper stage. However, in Germany several practical steps are being taken. Through the efforts of Dr. Foerster several new ships have been fitted with inclining tanks, the filling of which produces a heel from which, from particulars supplied to the ship's officer, the metacentric height may be easily calculated or read off from curves. Ships' officers are not usually credited with great arithmetical ability, and thus many attempts have been made to present stability information in a quasi-mechanical form. One of the most recent attempts is an instrument invented by Dr. Kempf, Director of the Hamburg Tank. Briefly, it consists of two distinct portions, one of which consists of a frame into which is placed a card upon which curves of righting arm for each angle are plotted horizontally to a vertical scale of draught. The sides of the frame form the draught scale and the values of righting arm are read off by means of a horizontal cursor sliding along these draught scales. These righting arms refer to a position of centre of gravity situated on the centre line at the keel. To correct for position of centre of gravity the second portion is made use of. Knowing the metacentric height, the modification to the righting arm is mechanically obtained. The instrument is easy to use, and it is pointed out that if on each voyage the instrument were used to construct the vessel's statical stability curve and full reports of the vessel's behaviour kept, then valuable information would be available for future designing purposes.—*W. Allner, Hansa*, April 28, 1923; p. 441; 2 pp.; 2 ff.

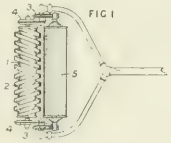
RECENT PATENTS RELATING TO MARINE ENGINEERING.

Compiled by E. P. Alexander & Sons, Chartered Patent Agents, 306, High Holborn, London, W.C.

No. 193,403. Cleaning Ships' Hulls.

Arentz, H., 28, *Lemchesvej, Hellerup, Denmark. February* 14, 1923, *No.* 4,364. *Convention date, February* 16, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Class* 113 (i).]

In machines for cleaning ships' hulls of the kind comprising a turbine-driven rotating brush 2 formed on the exterior of a cylindrical air container 1 serving as a buoyancy member, the size of the container and hence of the brush is kept within reasonable limits by the provision of an auxiliary buoyancy chamber 5. The turbines 3 driving the brush are arranged at either end of the spindle, and the pressure water is discharged to produce a reaction pressing the brush against its work. The gearings driving the brush are enclosed in casings 4 serving as supports for the turbines, and are provided with fittings for branch pipes through which pressure water is pumped to the turbines.



No. 193,507. Turbines.

English Electric Company Limited, and Seewer, P. W., 28, *Kingsway, London. November* 25, 1921, *No.* 31,632. [*Class* 110 (iii).]

In hydraulic turbines of the impulse type, the runner 1 has a central sharp edge 3 and is shaped as shown, the buckets 5 being formed separately and cast into the concave surfaces of the runner. The edges 8 of the buckets do not extend radially outward beyond the edge 3 of the runner until they have extended laterally from this edge to a distance equal to the radius of the full jet 11. The pitch of the buckets is not greater than the diameter of the full jet, and the bottom of the bowl of each bucket lies inside the plane of the outer edge 10 of the next bucket. Supporting rings 13 are cast round the outer peripheries of the buckets. Fig. 4 shows a modified form of bucket 5 with shaped reinforcing rings 13. The runner may have buckets on one side only of the edge 3.

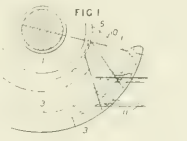


FIG.2.

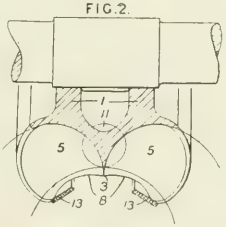
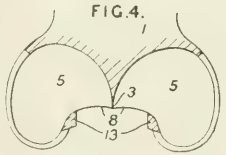


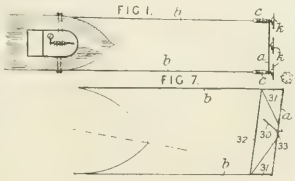
FIG.4.



No. 195,162. Automatic Pilots for Ships.

Shelly, S., 1925, *North* 32*nd Street, and Moyerman, S.*, 1113, *Lincoln Building, Broad Street, both in Philadelphia, U.S.A. December* 23, 1921, *No.* 34666. [*Classes* 9 (ii) and 113 (i).]

Apparatus for protecting ships from collision or for preventing them from running into submerged obstructions, such as rocks or mines, comprises a flat submerged advance guard *a*, maintained in a plane parallel with a transverse plane of the ship by leashes *b*, maintained in different horizontal and vertical planes, and kept in tension by motors *c* supplied with power from the parent ship. Indicating devices *k* are mounted at the corners and other points on the guard, and on impact with an object, the particular indicator struck is pressed back to close an electric circuit and operate a bell or electric light on an indicator board on the ship. By leading the bridles *b* to either side of the ship and maintaining them in tension, the guard is caused to shift its course into that of the ship. This end may also be achieved by yoking the bridles *b* to arms 33, Fig. 7, on a centrally-disposed rudder 30, the guard being further furnished with links 31 connected to an articulated parallelopipedon 32. Specification 21007*/14 is referred to.

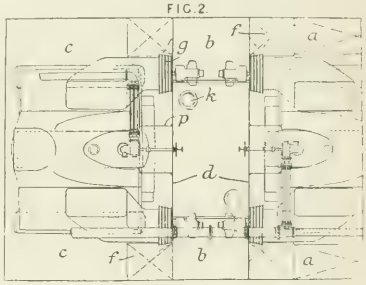


No. 195,197. Ships.

Thornycroft & Co. Ltd., J. I., Thornycroft House, Smith Square, Westminster, and Mackie, R., Woolston Works, Woolston, Hampshire. January 14, 1922, *No.* 1295. [*Classes* 113 (ii) *and* 137.]

Destroyers; stokeholds.—To enable the personnel of naval craft to carry out their duties without the aid of gas masks when the vessel enters or is enveloped in a gas cloud, certain compartments to which air is supplied in a purified condition are isolated from the remainder of the vessel, and are reserved for the use of the crew. The stokehold *b* is isolated from the adjacent boiler rooms *a*, *c* by bulkheads *d* fitted with air locks *f*. The forced draught for the boiler rooms is effected by geared turbo-fans *g*, and the air for the stokehold is drawn by fans or through ventilators *k* and a box containing purifying substances. The parts in the boiler room which require manipulation can be operated from the stokehold by extension spindles passing through the bulkhead, or the bulkhead may be recessed as at *p* to allow access to requisite parts.

FIG.2.

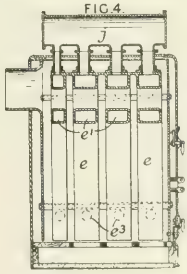


No. 195,202. Sectional steam boilers.

Allin, H. P., and Bruster, O., 4, *Lloyd's Avenue, London. January* 21, 1922, *No.* 1863. [*Class* 123 (ii).]

A number of sections *e* of substantially ∩-shape are arranged so as to be exposed on all sides to the combustion gases. Cross water passages e^1 are arranged in their upper parts. An externally disposed header *j* is connected separately to each section, and two other external headers, one on each side, are connected to the bottom ends of the legs of the ∩, at e^3.

FIG.4.

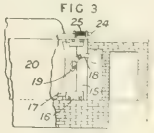


No. 195,212. Boiler flues.

Crawshaw, E., Mill Road, Walmersley, and Stuart, W., 7, *Lime Grove, Limefield, both in Bury. January* 30, 1922, *No.* 2672. [*Class* 123 (ii).]

Expansion joints between steam boilers and their flue walls comprise two vertical sliding plates in cavities in the walls, which are maintained against the boiler by weighted levers in combination with an upper horizontal sliding plate maintained in contact by springs. Each vertical plate 17 moves in guides 16 fixed to a plate 15 which also carries the weighted lever 18. The lever has a pin and slot connection 19 with the plate 17 and holds it against the boiler. A top plate 20 carries guides for a horizontal

FIG 3

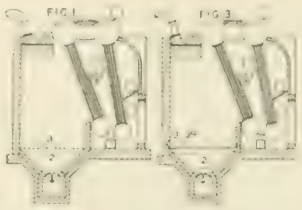


[illegible] plate which is held in contact by springs 25 on guide pins 24.

No. 195,247. Furnaces.

[illegible] *Fire Prevention Engineering, Ltd.*, 47, Victoria Street, Westminster. *March* 2, 1922. *No.* 6213. [*Class* 51 (i).]

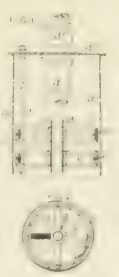
In a pulverised fuel furnace, the residues of combustion before entering the ash-pit 2, Fig. 1, pass through a tube of air, steam or water delivered under pressure from perforated pipes 3. In the modification shown in Fig. 3, the air, steam or water is delivered through a slot or slots 3 in a duct 3 of flattened oval cross-section arranged in the furnace wall immediately above the ash-pit 2.



No. 195,266. Electric condensers.

Goodman, W. H., 94, *Addison Road, Kensington, and Dubilier Condenser Company* (1921), *Ltd., Ducon Works, Goldhawk Road, Shepherd's Bush, both in London. March* 22, 1922. *No.* 8381. [*Class* 38 (iii).]

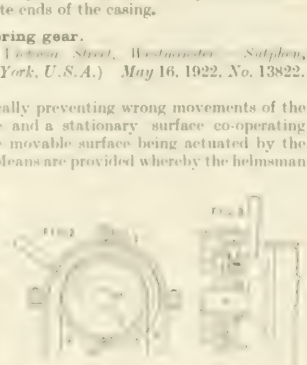
A condenser comprises sets of plates or units arranged radially around a centre or axis, each set comprising a number of plates or units mounted together in block form and arranged substantially tangential to concentric circles. The plates or units of high potential are preferably arranged adjacent the axis, those of low potential being arranged adjacent the periphery of the condenser. In the construction shown, a number of rings *c* are mounted co-axially on four rods *b* carried by the lid a^1 of the casing *a*, these rods forming low-potential terminals in parallel. The high-potential terminal is formed by an axial tube *d*, the tube and the high-potential lead being insulated from the casing by sleeves d^1, d^2. The tube *d* carries a number of polygonal collars c^1 in the plane of the rings *c*, the condenser elements *e* being clamped between the faces of the collars and corresponding faces c^2 on the rings *c* by screws c^3. In place of the collars, independent blocks adapted to bed against the tube *d* may be provided for each of the sets of elements *e*. The rods *b* and rings *c* may be replaced by a frame which may be suspended within the casing or carried by its lower end, or the casing itself may be provided with internal seatings for the sets of condenser elements. In a modification, the supports at the outer ends of the condenser sets are dispensed with, and the condenser elements are threaded on rods extending radially at right angles to the axis of the casing. The high and low potential leads may be arranged at opposite ends of the casing.



No. 195,299. Ships' steering gear.

[illegible] *Lawson Street, Westminster, Sutphen, E. W., 52, Broadway, New York, U.S.A.*) *May* 16, 1922. *No.* 13822. [*Class* 114.]

Mechanism for automatically preventing wrong movements of the helm comprises a movable and a stationary surface co-operating with a wedging device, the movable surface being actuated by the signal transmitting gear. Means are provided whereby the helmsman may cut out this control [illegible] keyed a disc 7, and there- [illegible] member 11, Figs. 2, 3, [illegible] which engages a pin 15, [illegible] prevented from rotating. The opposed faces 12, 13 of the discs 7, 11 form a wedge race for a ball 19. The eccentric

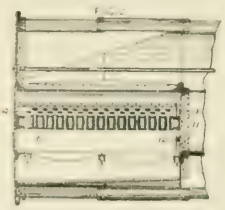


On the actuation of the cable 4, the disc 11 takes an eccentric position relative to the disc 7 and the hand-wheel can only be turned, without wedging, in the desired direction. The wedge races may be shaped to give four-point contact with the ball, or the surfaces of the races and wedge may be ribbed or toothed. In a modification, wedge-shaped members are slidingly attached to one of the discs. In a further modification, a train of wheels carried by the eccentric disc brings a locking pinion into engagement with teeth on the shaft disc. Rotation in the correct direction acts to disengage the locking pinion, but in the wrong direction the pinion is held hard.

No. 195,304. Steam boilers.

Francis, T. H., 1, *Millfield Villas, Pontypridd, Glamorgan. June* 7, 1922. *No.* 15776. *Classes* 51 (i) *and* 123 (iii).]

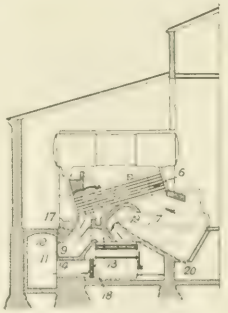
A furnace crown having cross fire-tubes is so arranged within a boiler furnace that the gases pass partly under the crown directly into the boiler flue and partly through the cross fire-tubes and then into the flue. The crown 4 in a furnace of a Lancashire boiler consists of a flat, longitudinal, open-sided water chamber which is formed by two parallel plates secured obliquely across the furnace. The chamber is traversed by fire-tubes 6. The bridge 11 has an aperture 12, allowing part of the gases to pass beneath the crown directly into the flue 9.



No. 195,329. Furnaces, &c.

Aktiebolaget Ljungströms Angturbin, 2, *Arsenalsgatan, Stockholm. September* 11, 1922. *No.* 24591. *Convention date, July* 1. [*Class* 51 (i).]

An air-preheater 13 of the rotating and regenerative type is mounted below a boiler system 6 in the space between the ash-pit 20 and chimney flue 10. Hot gases from the furnace 7 pass over the boiler tubes and thence through the channel 9, preheater 13 and channel 14 to the flue 10. Heated air passes by the damper 19 to below the furnace 7. Dampers 11, 17, 18 control the flow of the furnace gases. The walls forming the air and waste gas passages are part of the brickwork of the plant. In a plant comprising two or more boilers the preheater may be divided into two or more parts.



No. 195,354. Turbo-electric generators : pumps.

Huguenin, A., 19, *Hardturmstrasse, Zurich, Switzerland. February* 8, 1923. *No.* 3844. *Convention date, March* 21, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act* (*Classes* 35, 38 (iv), *and* 110 (iii).]

A turbo-electric generator unit intended for occasional use as an electrically-driven pump unit is arranged so that the turbine element runs at a higher speed when used as a pump than when used as a motor. This may be obtained by the use of change-speed gears between the two elements, and when the electric element is an alternating current machine, its speed can be varied by altering the number of effective poles or by a cascade arrangement.

OFFICIAL NOTICES.

THE [illegible] Patents No. 15,58, relating to Internal Combustion [illegible] [illegible] [illegible]

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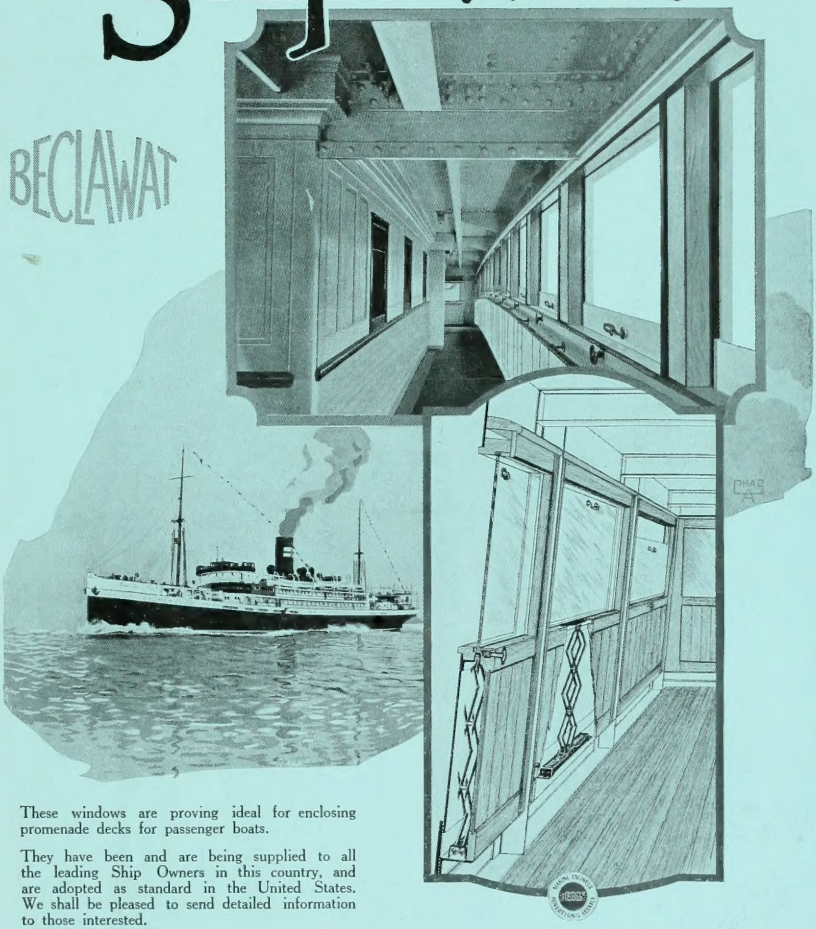

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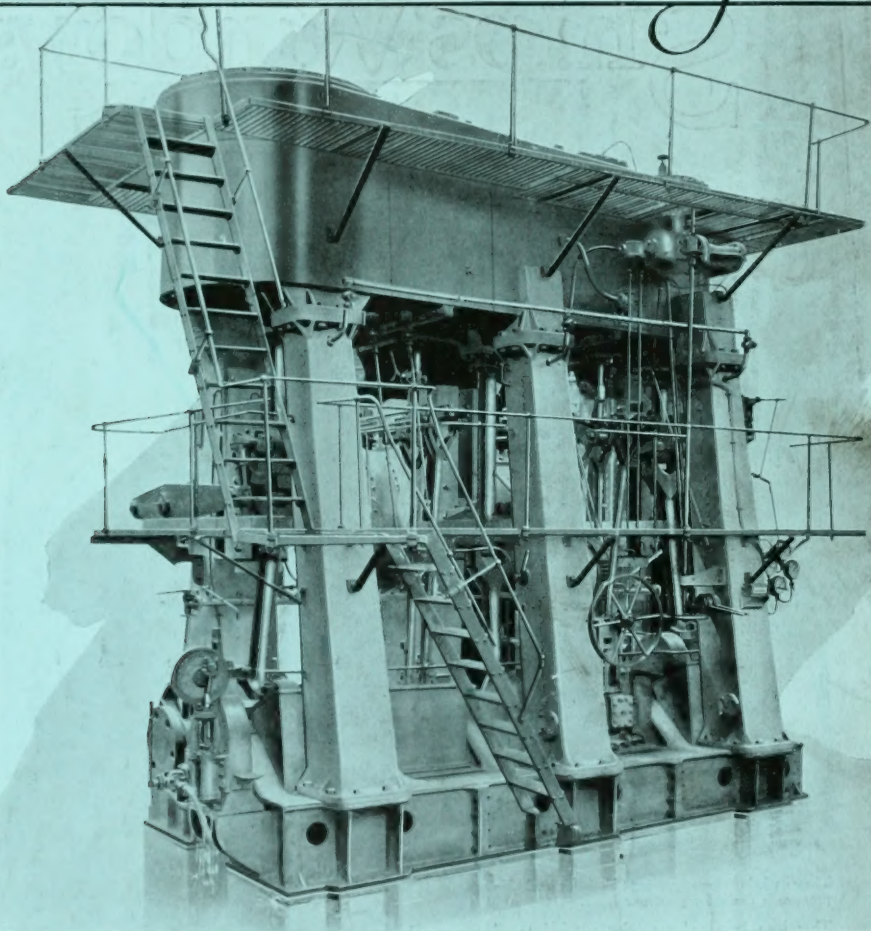

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